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FINAL REPORT

PRELIMINARY ANALYSIS OF A FLEXIBLE INSTRUMENT MOUNT FOR LARGE INSTRUMENTS ON THE SPACE SHUTTLE

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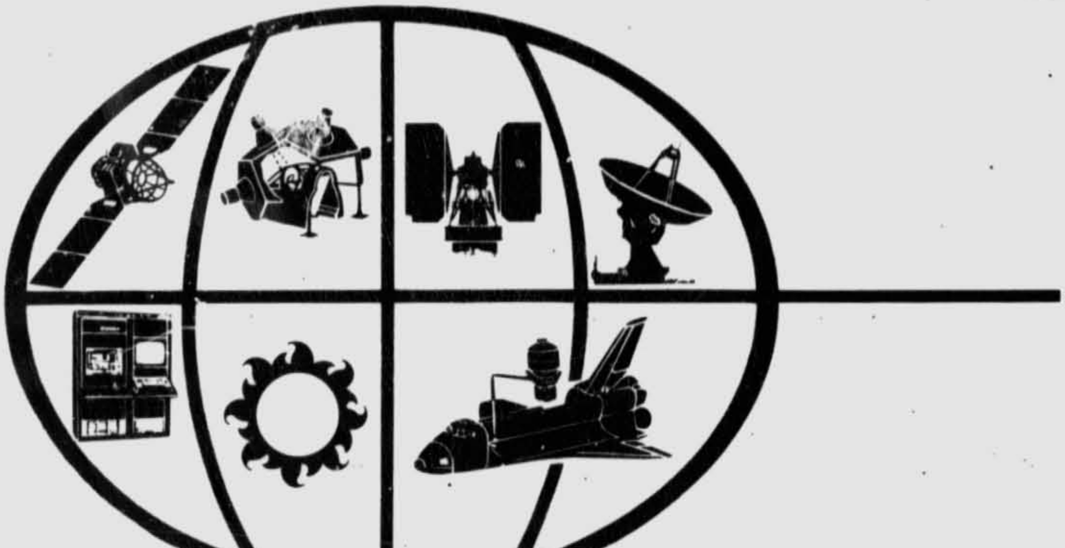

space division

GENERAL ELECTRIC

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PRELIMINARY ANALYSIS OF A FLEXIBLE INSTRUMENT MOUNT FOR LARGE INSTRUMENTS ON THE SPACE SHUTTLE

FINAL REPORT

20 DECEMBER 1978

Prepared For

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
GODDARD SPACE FLIGHT CENTER
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1. INTRODUCTION

This document constitutes the Final Report of a preliminary analysis of a Flexible Instrument Mount (FIM) for large instruments on the Space Shuttle. The analysis was carried out by the Space Division of the General Electric Company under Contract No. NAS 5-23412 for the NASA/GSFC. The overall objective of the study was to identify and define promising concepts for pointing instruments while in orbit, with weights up to 2000 Kg and dimensions of 2-3 meters.

With the advent of the Space Shuttle and Spacelab, a large number of science and technology payloads and experiments will be flown in near earth orbit on a regular basis between 1980 and 1990. Many planned and conceivable instruments will require some type of pointing, e.g. astronomy, astrophysics, solar physics, AMPS-type and earth viewing instrument classes. Pointing requirements will range from high accuracy arc second pointing or better to coarse pointing of the accuracy provided by the Shuttle Orbiter. Several highly accurate fine pointing systems are either under development or are considered for development e.g. the European IPS, and NASA's SIPS and ASPS.

Instruments satisfied with the Orbiter pointing capability can be hard mounted into the Orbiter cargo bay or onto a Spacelab pallet. However, significant mission flexibility would be gained if instruments requiring Orbiter pointing could be mounted on a gimballed mount which would allow offset pointing with Orbiter accuracy. For instance, it has to be expected that on many planned Spacelab missions more than one instrument could fly requiring Orbiter pointing.

If one such instrument would be mounted on a gimballed platform, two instruments could be operated simultaneously looking at different targets/areas with Orbiter pointing accuracy. This, of course, would significantly increase on-orbit timelining flexibility, increase scientific return and contribute to improve cost effectiveness of Shuttle payloads and operations. The flexible instrument mount described in this report fulfills the requirements for off-set pointing of large instruments to be flown on Spacelab missions.

After careful evaluation of several options available to meet the requirements of this flexible instrument mount, a mechanical concept was selected that can accommodate a set class of scientific instruments such as the LAMAR X-ray experiment with 24 LAMAR telescopes. The payload size was constrained by a desire to keep the FIM within the bounds of one Spacelab pallet and within the Orbiter payload envelope to preclude jettisoning the payload in case of manfunction. Thus, the payload size that can be accommodated is 2.25M diameter and 2.5M in length.

The basic FIM structure uses a classic yoke assembly with a bottom ring to achieve the gimballing in azimuth and elevation and a 60° (half angle) field of view. This structure is attached to the pallet at six hard point locations. The weight of the structure was not a prime design parameter. The structural members are sized to achieve a stable mount capable of accommodating heavy payloads with small structure deflections.

The drive system for controlling the azimuth and elevation motions is a torquer motor and a speed reducer designed by Compudrive Corp. The bearing for the azimuth drive is a large diameter machinery bearing (supplied by Rotek, Inc.) built ruggedly to withstand the large landing loads of the Shuttle.

Position indication of the drive system is achieved with use of an encoder (a multiturn potentiometer), with a read-out accuracy of 0.1^{o}.

This report is organized to the tasks conducted which lead to the identification of a feasible, cost effective FIM concept:

Section No.	Task	Task No.
2	Requirements Definition	1
3	Concept Development	2
4	Concept Evaluation	3
5	Engineering Definition	4
6	FIM Development Plan	5

Reference APPENDIX D EXECUTIVE SUMMARY for FIM Vu-Graphs.

2. REQUIREMENTS DEFINITION (TASK 1)

A requirements analysis was carried out to derive a set of design groundrules and constraints and to identify trade-off's in areas of conflicting requirements.

The system level requirements in the Statement of Work (PCN 420-66984) served as the starting point. Descriptions of several proposed instruments, representative of potential Spacelab experiment classes to be accommodated by FIM, were used to further refine and interpret the SOW requirements. The instruments identified by GSFC were a Gamma Ray Spectrometer, a Medium Energy Gamma Ray Spectrometer and a Large Area X-Ray Telescope (LAMAR).

Shuttle/Spacelab interfaces and operational requirements were extracted from the Spacelab Payload Accommodation Handbook (SPAH, Issue No. 1, Rev. No. 0) and SPAH Appendix B, Structure Interface Definition.

Some important general guidelines and constraints were identified and agreed upon during an orientation meeting at GSFC at the beginning of the study:

- The development of a <u>low cost</u> FIM concept was emphasized as a very important objective of the study.
- In view of current Shuttle charging policies for payloads it was recognized that the overall length in the Orbiter cargo bay of the FIM/experiment should be minimized.
- A standard Spacelab pallet should be used for FIM mounting to reduce the uncertainty of interface engineering, fabrication, and integration costs for an independent mount.
- Penetration of the Orbiter cargo bay payload envelope should be avoided during all flight phases to eliminate jettison requirements.

- FIM should be designed to allow 1-g testing without extensive ground support equipment.

The above stated guidelines were subsequently adopted as overall design groundrules.

2.1 PHYSICAL ACCOMMODATION REQUIREMENTS

Within the above listed constraints it was stated as a goal that FIM accommodate

- cylindrical instruments, 1.2 m to 2 m diameter, 3 m long
- rectangular instruments, 2 m x 2 m x 3 m long

Instrument weight to be accommodated was stated as

- instrument weight, 950 kg to 2000 kg.

In addition, it was assumed that the structural interface of the instrument to be mounted to FIM would be a flange on the instrument, on a plane perpendicular to the line-of-sight at the equator and near the cg of the instrument. The cg of the instrument was assumed to be within 0.25 m of the geometric center of the instrument.

In addition to accommodating the overall instrument envelopes listed above, it was also required that a thermal cannister enclosing the instrument be accommodated. Typical thermal cannisters would add a few centimeters to the dimensions of an instrument envelope.

During the analysis of various potential FIM concepts it became apparent that the 3 m instrument length requirement was difficult to accommodate within the overall constraints and requirements, i.e. without violating

- the pallet-mounted FIM concept
- staying within the Orbiter cargo bay dynamic envelope
- maintaining the full 60° pointing range capability

The 3m length requirement was driven by the LAMAR instrument. An investigation of the LAMAR requirements (Ref. 1) and a discussion with Mr. Robert Rasche, LAMAR project engineer at the Smithsonian Astrophysical Observatory resulted in the following:

As far as sensitivity is concerned, an assembly of 24 LAMAR telescopes is so sensitive that adequate data from even weak X-ray sources can be obtained if pointing is off-axis by as much as 0.5 degrees (i.e. Orbiter pointing accuracy). The LAMAR in any configuration requires a star tracker/camera for after the fact aspect determination. If this aspect data is used in real time by the Orbiter attitude control system, adequate pointing can be obtained without a pointing system. Thus, if LAMAR was mounted to FIM it could be pointed at target areas with Orbiter pointing accuracy, simultaneous with other experiments hard mounted to a Spacelab pallet requiring different view directions.

It was determined in a discussion with Mr. Rasche that some flexibility exists in adjusting the overall LAMAR length. The 2.8 m required in Ref. 1 (without thermal precollimator) can be reduced to 2.6 m by repackaging the support systems for the imaging proportional counter in the focal plane. A later version of LAMAR shown in a NASA Headquarters presentation in May, 1978 shows a Spacelab pallet mounted LAMAR that requires even less than 2.6 m length, including a thermal precollimator. Based on this it was decided to modify the requirement for accommodating a package of 2m x 3m x 3m, to accommodating a minimum of 24 LAMAR telescopes of at least 2.5m length. The requirement for cylindrical payloads was not changed.

Ref. 1: Experiment Definition and Integration Studies for the Accommodation of an Aries X-ray Telescope Payload on Spacelab/Shuttle. Final Report for NASA Contract NAS 5-23685. Smithsonian Astrophysical Observatory.

2.2 FUNCTIONAL ACCOMMODATION REQUIREMENTS

The FIM functional performance requirements are determined by the pointing and alignment requirements of instruments to be flown on FIM. The support services requirements are derived from FIM and instrument requirements for power, thermal control and command and data management.

POINTING REQUIREMENTS

The FIM pointing requirements are as follows:

- Pointing Range (FOV) - anywhere in a 60° half angle cone around the instrument center line, parallel to the Orbiter Z (yaw) axis.
- Pointing Accuracy - 1° relative to the base of FIM (i.e., the pallet).
- Encoder Readout - 0.1°
- Pointing Stability - not applicable since FIM is not an active pointing system. The pointing stability offered to FIM mounted instruments is the Orbiter pointing stability (see SPAH, Section 2.4).
- Offset Slewing Capability - maximum slew rate of 120° in one minute (goal); minimum slewing rate is 40° in one minute.
- Frequency of offset pointing maneuver - as often as every ten minutes.

With respect to pointing accuracy, the following needs to be understood:

1. The Orbiter can point any vector defined in the Orbiter Navigation Base (this is a structural reference in the Orbiter nose section) to within $\pm0.5°$, including earth targets.

2. The alignment of pallets in the Orbiter cargo bay with respect to the Navigation Base can be off by as much as 2° to 5°, and can change on-orbit as a function of the thermal environment which can cause Orbiter torsions.

3. Therefore, in order to have accurate aspect information and to make full use of Orbiter pointing capabilities, the payload must provide an aspect sensor system. This sensor system can interface with the Orbiter Guidance, Navigation and Control System if payload aspect information is to be used to point the Orbiter. No interface with the Orbiter is necessary if only after the fact aspect information is required.

REQUIREMENTS FOR SPACELAB ELECTRICAL SERVICES

For the operation of payloads and payload support systems like FIM, the Spacelab offers electrical power and command and data management services on the pallet at standard interface locations.

ELECTRICAL POWER

FIM shall use Spacelab provided electrical power. As a goal, power consumption shall be kept to a minimum (below 100 W).

COMMAND DATA MANAGEMENT

FIM shall use the Spacelab command and data handling system (augmented by its own microprocessor if necessary) to allow remote control by the Payload Specialist or from the ground.

ELECTRICAL SERVICES TO FIM MOUNTED INSTRUMENTS

FIM shall provide access to the Spacelab EPDS and CDMS services for FIM mounted instruments by means of a power and data harness, running across the FIM gimbals from the instrument support ring to connector brackets at the FIM base. As a minimum, the following services shall be provided:

EPDS	- 3 buses for primary DC 200 W max. cont., 350 W peak each
	- 1 bus for experiment essential power 100 W max. cont.
	- 1 dual redundant bus for emergency power 50 W max. cont. each
CDMS	- wiring for 3 experiment RAU's
	- 6 TSP, 125 Ohm impedance for 3 HRM channels up to 16 MB/S
	- 2 TSP, 75 Ohm impedance for 1 CCTV channel plus sync
	- 1 TSP, 75 Ohm impedance for 14.5 MHz analog channel
	- 10 pairs flat conductors shielded for caution and warning and other payload functions.

THERMAL CONTROL REQUIREMENTS

The FIM concept shall be compatible with thermal control techniques (active and passive) that allow FIM bulk temperatures to be maintained at 20°C ±20°C.

In addition, the FIM concept shall allow that thermal canisters for instrument thermal control can be accommodated. Otherwise instrument thermal control is the responsibility of the instrument mounted to FIM. It was specifically agreed that FIM mounted instruments would not require that the pallet freon cooling loop be brought across FIM gimbals.

2.3 SHUTTLE/SPACELAB INTERFACE REQUIREMENTS

Stipulated by the RFP requirements, and in order to cover the complete spectrums of potential FIM configurations, two basic ways of mounting FIM into the Orbiter cargo bay were initially considered:

1. FIM mounted directly into the Orbiter cargo bay, using the primary Orbiter attachment points on the cargo bay sills and the cargo bay keel fittings.

2. FIM mounted to a Spacelab pallet, using the pallet hardpoints.

In both cases it is possible to fly the FIM (and experiments mounted to it) as part of a Spacelab mission or independent of Spacelab (e.g. missions of opportunity). This is an important distinction since Spacelab provides a number of services to its payload that the Orbiter does not provide, e.g. conditioned power, cold plate cooling, a command and data management system with a dedicated experiment computer, a high rate data acquisition system (multiplexer and recorder), and controls and displays (CRT and keyboard).

Flying on non-Spacelab missions, therefore, would require that the FIM provide the subsystems necessary to condition the raw Orbiter provided resources (power, cooling, Orbiter avionics services) for its own use of FIM mounted instruments.

Since the FIM has to be capable of accommodating large and heavy experiments, the FIM concept that was developed had to take into account the way the FIM instruments are mounted and structurally supported, especially during the critical launch and descent phases of a Shuttle flight. Again, two basic concepts had to be considered:

1. The FIM and its payload are structurally decoupled during launch and descent (similar to the IPS concept), and the FIM payload is mounted with a special payload clamp assembly
   a) to a Spacelab pallet, or
   b) directly into the Orbiter cargo bay

2. The FIM and payload are attached to each other with the FIM supporting all flight loads.

INTERFACE IDENTIFICATION

The first step in defining the FIM Shuttle/Spacelab interface requirements was the identification of all physical and functional interfaces for the various concepts identified above.

Table 2-1 represents the physical and functional interfaces as a function of:

1. The FIM mounting concept:
   a) Orbiter cargo bay (Shuttle)
   b) Spacelab Pallet (Pallet)

2. The mission mode:
   a) Spacelab missions
   b) Other Shuttle missions

3. The FIM/payload attachment concept during launch and descent:
   a) FIM/payload attached
   b) FIM/payload decoupled

Also identified are the interfaces the FIM experiments require. Depending on the FIM concept and mission mode, these interfaces are functionally directly between the experiments and Shuttle/Spacelab (physically they will be located on the FIM) or directly between the experiments and FIM provided subsystems.

It was realized early in the study that only a pallet mounted FIM would allow a low cost approach, and mounting FIM to a Spacelab pallet was adopted as a design groundrule. In addition, it was agreed that only Spacelab missions will be considered during this study.

TABLE 2-1. FIM (PAYLOAD) INTERFACE MATRIX

				SHUTTLE/ORBITER						SPACELAB					
CONCEPT	INTERFACES 1)	MISSION MODE 2)	FIM/PAYLOAD MOUNTING 3)	STRUCTURE	ELEC. POWER	THERMAL	AVIONICS	SOFTWARE	CAUTION & WARNING	STRUCTURE	ELEC. POWER	THERMAL	CDMS	SOFTWARE	CAUTION & WARNING
FIM/PAYLOAD DECOUPLED DURING LAUNCH & RE-ENTRY	FIM	SL	PAL					(X)	X	X	X	X	X	X	X
	PL		PAL						(X)	X	X	(X)	X	X	X
	FIM	SL	ORB	X		X		(X)	X		X	(X)	X	X	X
	PL		PAL						(X)	X	X	(X)	X	X	X
	FIM	SH	ORB	X	X	X	X	X	X						
	PL		ORB	X	[X]	[X]	[X]	X	(X)						
	FIM	SH	PAL		X	X	X	X	X						
	PL		PAL		[X]	X	[X]	X	(X)						
	FIM	SH	ORB	X	X	X	X	X	X						
	PL		PAL		[X]	[X]	[X]	X	(X)						
FIM/PAYLOAD ATTACHED	FIM	SL	PAL					(X)	X	X	X	X	X	X	X
	PL								(X)	[X]	X	(X)	X	X	X
	FIM	SL	ORB	X		X		(X)	X		X	(X)	X	X	X
	PL								(X)	[X]	X	(X)	X	X	X
	FIM	SH	PAL		X	X	X	X	X						
	PL			[X]	[X]	[X]	[X]	X	(X)						
	FIM	SH	ORB	X	X	X	X	X	X						
	PL			[X]	[X]	[X]	[X]	X	(X)						

1) FIM and payload (i.e., FIM experiments) interfaces are shown separately
2) SL means Spacelab missions, SH means other Shuttle missions
3) PAL means pallet mounted, ORB means Orbiter mounted
(X) means potential interface, [X] Interface with Shuttle/Orbiter or FIM provided subsystem

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It was also determined that a FIM concept could be developed that would accommodate all instrument flight loads, thereby eliminating the need for any instrument to pallet or Orbiter structural interfaces. This leads to a significant reduction in the number of interfaces that had to be considered, as shown in Table 2-2.

MECHANICAL INTERFACES

The primary structural interface between the FIM and the Spacelab pallet are the pallet hardpoints. The pallet was designed for a nominal load carrying capability of about 3000 kg, uniformly distributed within certain cg constraints. The actual load carrying capability is very much a function of the actual load distribution and the hardpoint utilization pattern.

For heavy instruments the FIM/instrument system can approach the nominal pallet capability. This requires eventually a FIM pallet coupled analysis which was above the scope of this study. In addition, a payload/pallet coupled analysis can currently only be carried out by ESA and their Spacelab contractor and by the Spacelab project office at NASA-MSFC.

In addition to the overall load carrying capability of the pallet, the load carrying capability of each hardpoint also has to be taken into account. Preliminary hardpoint loads introduced by FIM were estimated in Appendix A.

In order to accommodate varying instrument masses at minimum FIM/instrument weight, a requirement was formulated to design a modular, flexible FIM/pallet interface structure that can be configured for optimum hardpoint utilization.

Two problems are of particular concern for pallet/payload interface structures.

1. large hardpoint location tolerances between various pallets
2. pallet/hardpoint deflections induced by Orbiter and payload loads and thermally induced torsions.

TABLE 2-2 FIM (PAYLOAD) INTERFACE MATRIX

CONCEPT	INTERFACES 1)	MISSION MODE 2)	FIM/PAYLOAD MOUNTING 3)	SHUTTLE/ORBITER STRUCTURE	SHUTTLE/ORBITER ELEC. POWER	SHUTTLE/ORBITER THERMAL	SHUTTLE/ORBITER AVIONICS	SHUTTLE/ORBITER SOFTWARE	SHUTTLE/ORBITER CAUTION & WARNING	SPACELAB STRUCTURE	SPACELAB ELEC. POWER	SPACELAB THERMAL	SPACELAB CDMS	SPACELAB SOFTWARE	SPACELAB CAUTION & WARNING
FIM/PAYLOAD DECOUPLED DURING LAUNCH & RE-ENTRY	FIM	SL	PAL					(X)	X	X	X	X	X	X	X
	PL		PAL						(X)	X	X	(X)	X	X	X
	FIM	SL	ORB	X		X		(X)	X		X	(X)	X	X	X
	PL		PAL						(X)	X	X	(X)	X	X	X
	FIM	SH	ORB	X	X	X	X	X	X						
	PL		ORB	X	[X]	[X]	[X]	X	(X)						
	FIM	SH	PAL		X	X	X	X	X						
	PL		PAL		[X]	X	[X]	X	(X)						
	FIM	SH	ORB	X	X	X	X	X	X						
	PL		PAL		[X]	[X]	[X]	X	(X)						
FIM/PAYLOAD ATTACHED	FIM	SL	PAL		REMAINING INTERFACES			(X)	X	X	X	X	X	X	X
	PL								(X)	[X]	X	(X)	X	X	X
	FIM	SL	ORB	X		X		(X)	X		X	(X)	X	X	X
	PL								(X)	[X]	X	(X)	X	X	X
	FIM	SH	PAL		X	X	X	X	X						
	PL			[X]	[X]	[X]	[X]	X	(X)						
	FIM	SH	ORB	X	X	X	X	X	X						
	PL			[X]	[X]	[X]	[X]	X	(X)						

1) FIM and payload (i.e., FIM experiments) interfaces are shown separately
2) SL means Spacelab missions, SH means other Shuttle missions
3) PAL means pallet mounted, ORB means Orbiter mounted
(X) means potential interface, [X] Interface with Shuttle/Orbiter or FIM provided subsystem

With a limited number of pallets available in NASA's inventory it has to be expected that the FIM will fly on various pallets. The interface structure, therefore, has to be designed to accommodate the hardpoint location tolerances.

Pallet deflections can be accommodated to some extent by the pallet hardpoints. Analysis of pallet deflection will eventually be necessary to determine to what extent the FIM interface structure will be impacted by these deflections, and how they can be accommodated.

It should be pointed out that the European IPS is faced with very similar problems. An IPS/pallet analysis is currently in process and results of this should be secured for further FIM analysis.

In summary it is concluded that the FIM/pallet mechanical interface is critical. The FIM interface structure needs to be modular and flexible for optimum FIM/instrument load accommodation. The detailed interface design needs to take into account significant pallet hardpoint location tolerances and potentially large pallet deflections. A FIM/pallet coupled analysis needs to be conducted during the next engineering phase to finalize the interface structure design.

THERMAL INTERFACE

The severe temperature environment and temperature gradients, that any payload in the Orbiter cargo bay can encounter (depending on mission profile and Orbiter attitudes), will impact the thermal design of the FIM. The Spacelab pallet, as an example, can reach steady state temperatures of -150°C in the cold case (deep space viewing), and +120°C in the hot case (solar viewing). Significant temperature changes and graidents can result as a function of changing Orbiter attitudes.

On the other hand, FIM mounted experiments will require thermal control over a much narrower and benign temperature range. Instrument thermal control is primarily achieved by the instrument itself, e.g. through the use of a thermal cannister which isolates the instrument from the pallet environment.

In general it seems reasonable to formulate for FIM requirements similar to those for the IPS (See SPAH Section 4.8.5):

FIM thermal control shall allow continuous cold case operation, i.e., deep space viewing for astronomy/astrophysics type instrument. This seems reasonable since the necessary orbiter attitudes can be achieved for all β -angles.

Certain operational constraints shall be allowed for full solar illumination in order to maintain acceptable FIM component temperatures, e.g. limitations on operating timelines.

The FIM thermal control shall be achieved primarily by passive means, e.g. multilayer insulation and/or thermal coatings, and radiators and electrical heaters if necessary for critical FIM components.

Results of IPS thermal analysis and details of the IPS thermal design should be evaluated during the next phase of the FIM development.

ELECTRICAL POWER INTERFACES

On Spacelab, conditioned dc and ac power is available at the output of the electrical power distribution box which is located on each Spacelab pallet. This power shall be used, and further conditioned if necessary, for the FIM itself, and shall be routed across the FIM gimbals and made available to experiments as shown in Section 2.2.

Also, in case Shuttle/Spacelab safety critical functions are identified, the distribution and conditioning of Spacelab emergency and essential power has to be taken into consideration.

COMMAND AND DATA MANAGEMENT INTERFACES

The interface requirements established under this heading include the physical and functional interfaces with the Spacelab command and data management system and includes Orbiter and Spacelab caution and warning requirements. The instrument command and data management interfaces are of minor concern here since all the FIM has to provide are the interface connectors and harnesses to connect experiments to the Spacelab CDMS.

On Spacelab missions, the Spacelab CDMS is available to handle all FIM command, control, monitoring and data handling requirements. The CDMS allows automatic or on-board manned control from the Orbiter aft flight deck (or the Spacelab module), and it establishes through the Orbiter avionics and telemetry systems the link to the payload operations control center (POCC) for ground monitoring and control.

One basic decision that had to be made is whether the CDMS subsystem computer or experiment computer should be used for the FIM. The Spacelab IPS uses the Subsystem computer which has the advantage of reserving the experiment computer capability for experiment operation. The disadvantage is that the subsystem and experiment computer cannot directly communicate with each other, which can add operational and software complexity. The decision was made for the FIM to use the Spacelab experiment computer (potentially augmented by a dedicated FIM mini or micro-processor).

The FIM shall interface with the Spacelab CDMS through an experiment RAU mounted to the pallet. This can be the same RAU used for experiment control because of the large capacity of an experiment RAU.

Since it can be assumed that the FIM will be used on many pallet-only missions it is important to minimize or eliminate the need for dedicated FIM controls and displays. This is the result of severe limitations in the Orbiter aft flight deck for payload provided equipment (volume, panel size, power, cooling). Since the FIM will interface with the Spacelab CDMS even when using its own processor, the Spacelab CRT and keyboard in the AFD can be shared with experiments for FIM control.

OPERATIONAL REQUIREMENTS

The FIM shall be designed for an operational life of 10 years or 50 flights with periodic maintenance and replacement/refurbishment of critical components, which shall be identified.

Ground integration and checkout have to be spearated into two phases:

1. Pre-Level IV integration, i.e., instrument to FIM integration
2. FIM to Spacelab integration (Level IV through II).

Based on recent developments it is expected that all Spacelab/payload integration (Level IV through Level I) will be carried out at KSC. The question remains open if pre-Level IV integration will also be done at KSC. In any case, a pallet simulator will be required if pre-Level IV integration as defined above is required.

If FIM/instrument integration takes place after FIM/Pallet integration, a simple handling fixture for FIM might be sufficient for ground handling.

An important requirement for the FIM design is the need to test the FIM/instrument assembly in 1-g without extensive test fixtures.

The most significant flight operational requirement is the requirement to control the FIM both from the on-board payload specialist and remotely from the ground POCC.

2.4 FIM DESIGN REQUIREMENTS SUMMARY

The FIM requirements identified and discussed in the preceding sections are summarized in Table 2-3, along with the compliance of the selected FIM concept described in detail in Section 5 of this report.

Table 2-3. FIM Requirements Summary

PARAMETER	REQUIREMENT	COMMENTS	COMPLIANCE
<u>INSTRUMENT ACCOMMODATION</u> - Size	Cylinder: 1.2 to 2 m dia., 3 m long and 24 LAMAR Modules, 2.5 m long	Goal: 2m x 3m x 3m length not achievable within constraints	Cylinder: 2.25m dia. 2.5m long Complies
- Weight	950 kg to 2000 kg		Complies
- Structural Interface	Center Support Flange		Complies
- CG Offset	0.25 m		On-orbit: Complies Ground: 0.05m without GSE
<u>INSTRUMENT SERVICES</u>			
- Pointing Range	60° half angle cone around Instrument center line		Complies
- Pointing Accuracy	1° relative to FIM base (i.e., Spacelab pallet)		Complies
- Encoder Readout	0.1°		Complies
- Pointing Stability	Not Applicable	Orbiter pointing stability	Orbiter pointing stability
- Slew Rate	120°/min. maximum (Goal) 40°/min. minimum		Complies
- Slewing Maneuvers	Once every 10 minutes (max.)		Complies
- Electrical Power	Harness across gimbals to Spacelab EPDS (pallet EPDB)	Same capability as offered by IPS	Complies
- Command and Data Management	Harness across gimbals to Spacelab CDMS (pallet RAU)	Same capability as offered by IPS	Complies
- Thermal Control	Accommodate Thermal Canister	Instrument Responsibility No freon lines across gimbals.	Complies

Table 2-3. FIM Requirements Summary (Continued)

PARAMETER	REQUIREMENT	COMMENTS	COMPLIANCE
SHUTTLE/SPACELAB INTERFACES			
- Envelope	Stay within Orbiter cargo bay payload envelope at all times, stay on one pallet.	To avoid emergency jettison requirement	Complies
- Structural	Use pallet hardpoints		Complies
- Electrical Power	Use Spacelab EPDS (pallet EPDB) Consumption < 100 W		Complies: 70 W max.
- Command and Data Handling	Use Spacelab CDMS (pallet RAU)	Augment with dedicated processor if necessary	Complies
- Control	On-orbit payload specialist and remote from ground POCC		Complies
- Thermal	Controlled to $20^{o}C \pm 20^{o}C$ bulk temp.	Thermal insulation, radiators and heaters	Complies: Passive thermal control (radiators and heaters if necessary)
OPERATIONS			
- Design Life	10 years or 50 missions	With periodic mainteance with refurbishment	Complies: periodic maintenance and refurbishment
- Integration and Test	Allow 1-g testing without special GSE		Complies
- Flight Operations	On-orbit and ground control, minimize controls and displays in Orbiter AFC	Crucial for pallet-only missions	Complies

3. CONCEPT DEVELOPMENT (TASK 2)

The concept development task utilized the requirements definition to generate a number of candidate concepts and to perform the initial screening to identify the two or three most promising for further evaluation. This was accomplished by a configuration synthesis which identified candidate gimballing and mechanical approaches, interface analyses which supported the configuration synthesis, and the feasibility analyses which evaluated the concepts as formulated.

3.1 CONFIGURATION SYNTHESIS

Several alternative concepts for mounting and pointing a large instrument package within the confines of the Orbiter payload bay were generated. Each concept was developed to the extent needed for comparative evaluation with respect to specific requirements.

The basic alternatives available for the development of a FIN concept are as follows:

1. Axis Orientation
   a. Azimuth - Elevation
   b. Roll - Pitch
2. Shuttle Mechanical Interface
   a. Spacelab Pallet Mount
   b. Direct Orbiter Mount
3. Instrument Mechanical Interface
   a. Center Flange
   b. End Flange or Plate
4. Deployment
   a. Rotation only
   b. Translation
   c. Extendable support

Each potential FIM configuration can be defined by a combination of the above basic approaches.

NOTE: During the initial phase of the study non-pallet FIM concepts were established and evaluated for comparison purposes. These configurations were then dropped and only pallet-mounted concepts were further evaluated, primarily because of the significantly higher development effort and cost of FIM's providing their own support structure for Orbiter mounting.

POTENTIAL FIM CONFIGURATIONS

Four basic concepts and two additional derivatives were developed for a total of six FIM candidate configurations. They are shown in Figures 3-1 to 3-6:

- Concept #1: Azimuth - Elevation Axes, CG - Mount
- Concept #1: Roll - Pitch Axes, CG - Mount
- Concept #3: Azimuth - Elevation Axes, End - Mount
- Concept #4: Azimuth - Elevation Axes, End - Mount, Deployable
- Concept #2A: Roll - Pitch Axes, CG - Mount (Extended)
- Concept #4A: Azimuth - Elevation Axes, CG - Mount, Non-Pallet Mounted, Deployable

A brief discussion of each concept is given below.

CONCEPT #1 (A_z - EL)

Both gimbal axes can be made to go through the center of gravity of the instrument so that a balanced load is achieved. A large azimuth bearing is located below the instrument which tends to increase the overall hight of the FIM/instrument assembly. The az-el approach has the advantage of placing the instrument c.g. over the azimuth bearing centerline, providing a rigidly symmetrical unit. This and the c.g. mount allow, therefore, ground testing without the aide of a "zero-g" support kit.

In addition, Concept #1 makes good use of the available pallet volume. It also allows a relatively simple, symmetrical pallet/FIM interface structure for distributed load introduction into the pallet.

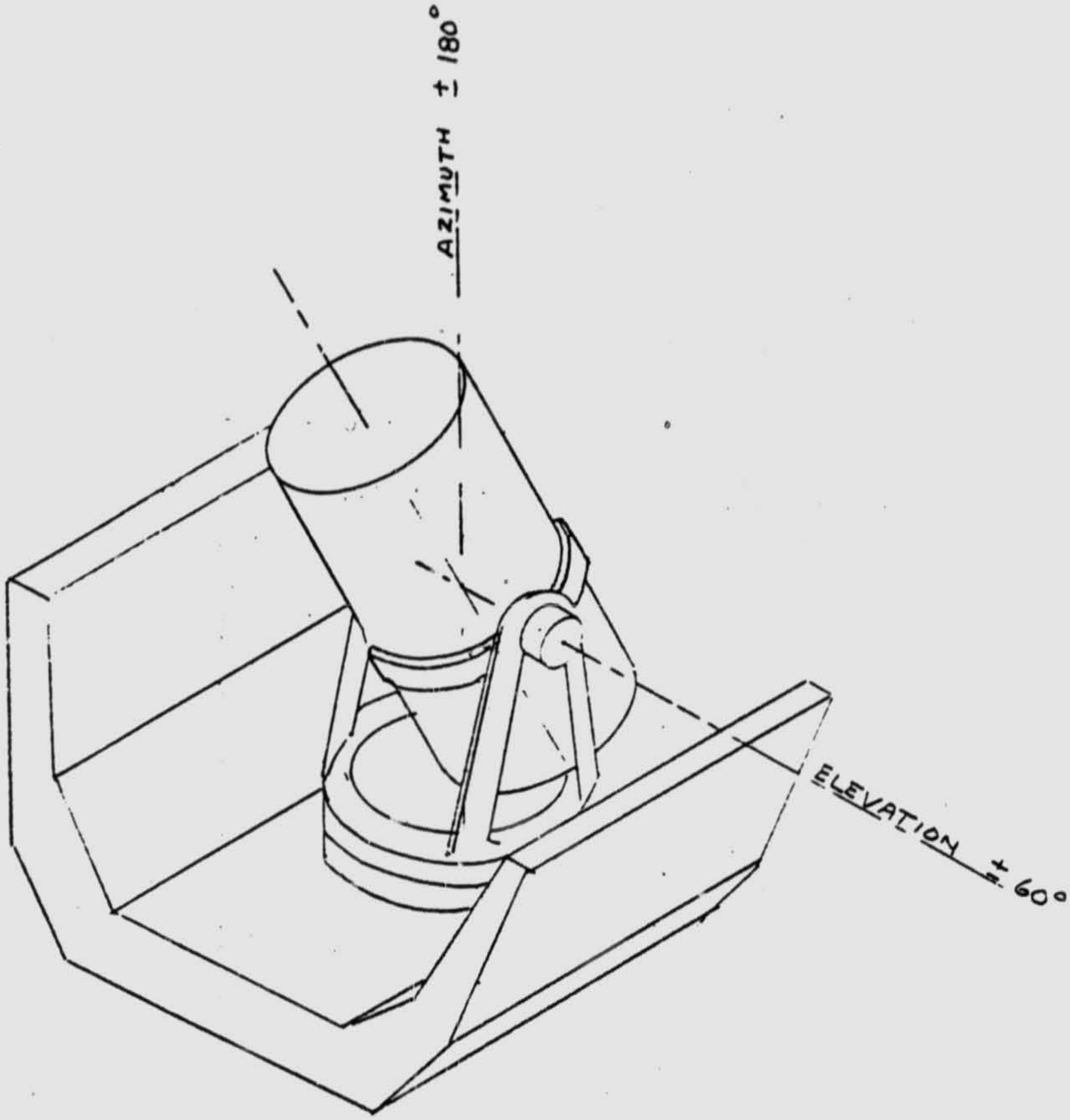

FIGURE 3-1 CONCEPT # 1 (AZIMUTH - ELEVATION)

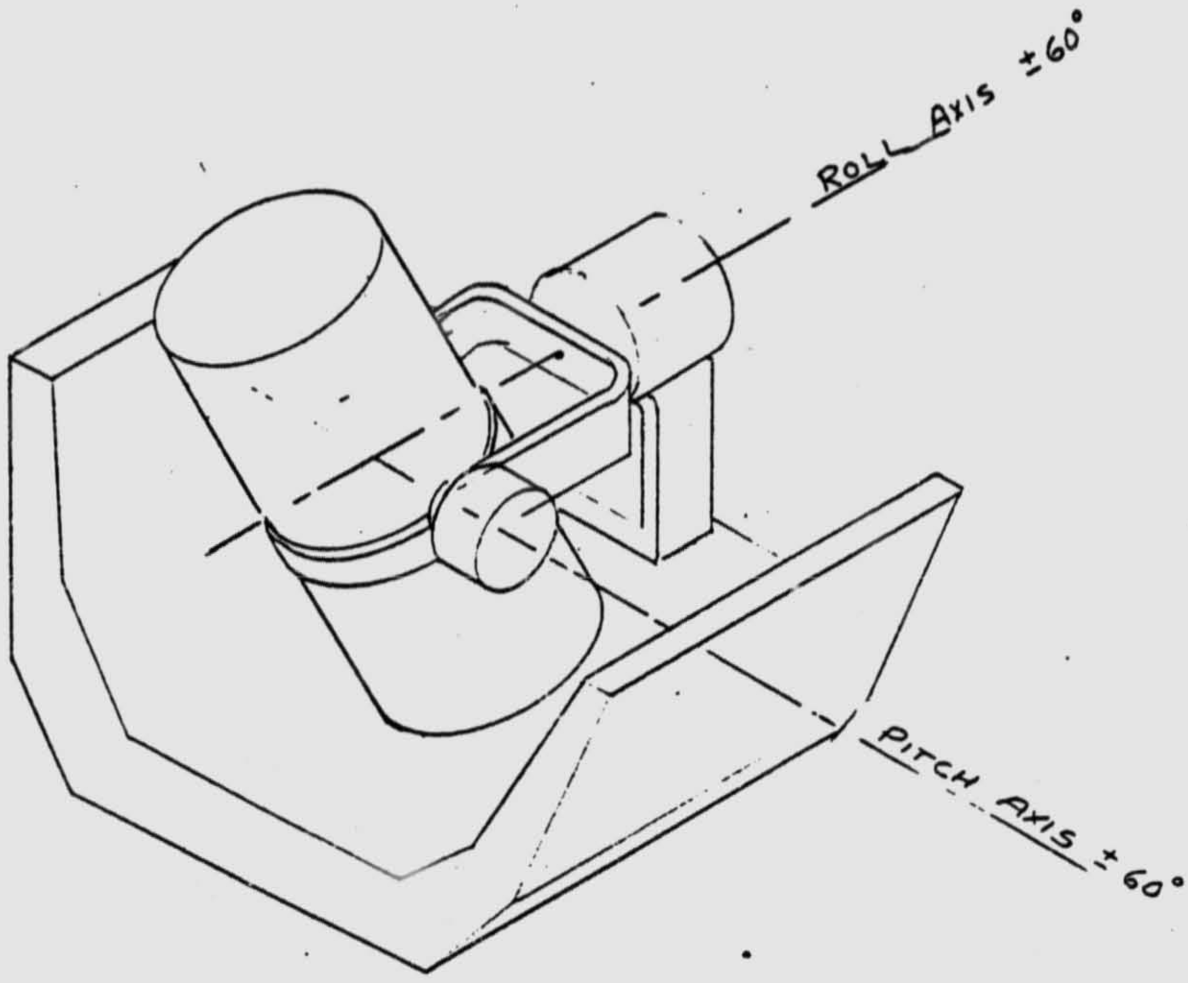

FIGURE 3-2 CONCEPT # 2 (ROLL-PITCH)

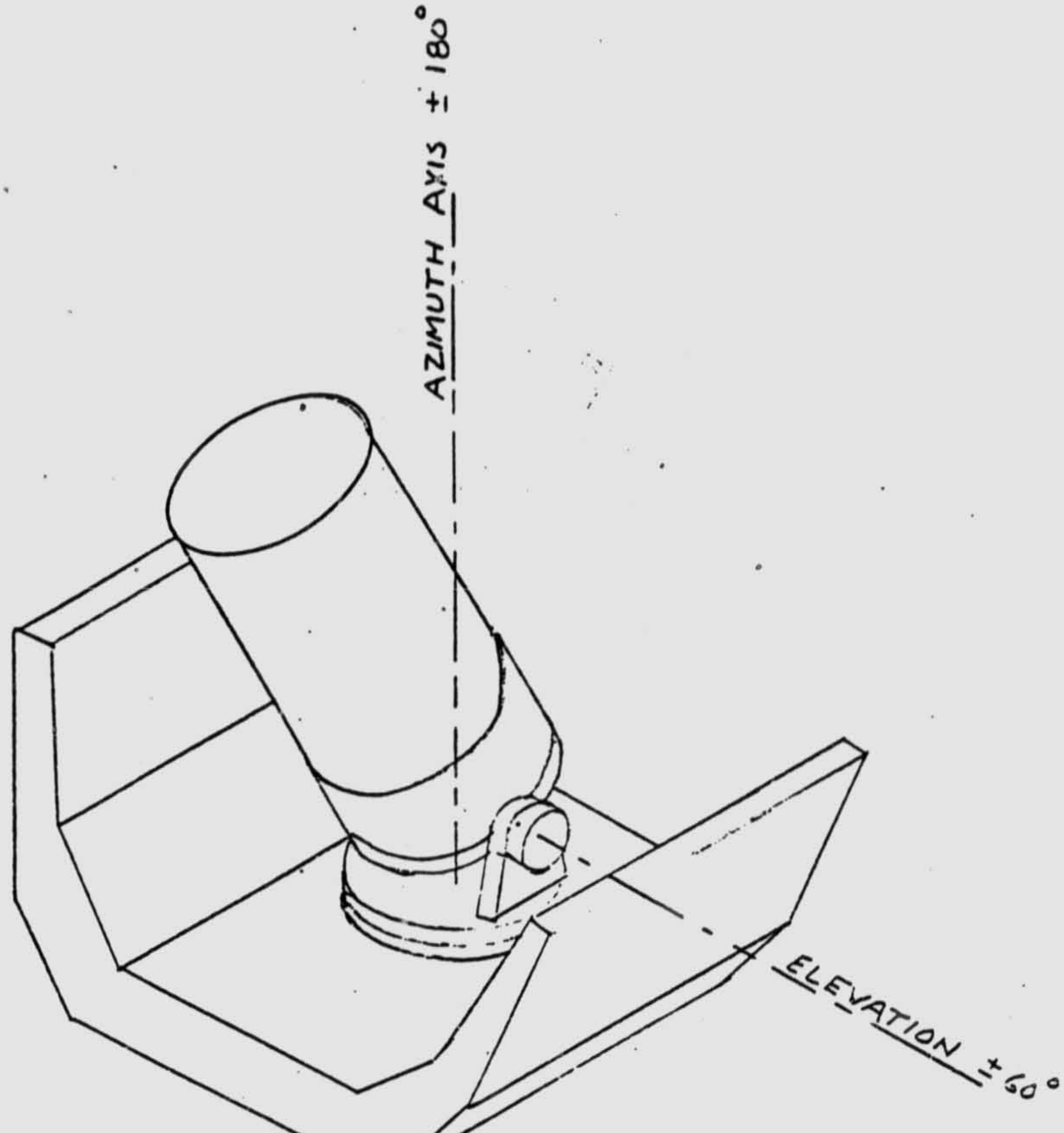

FIGURE 3-3 CONCEPT # 3 (END MOUNT)

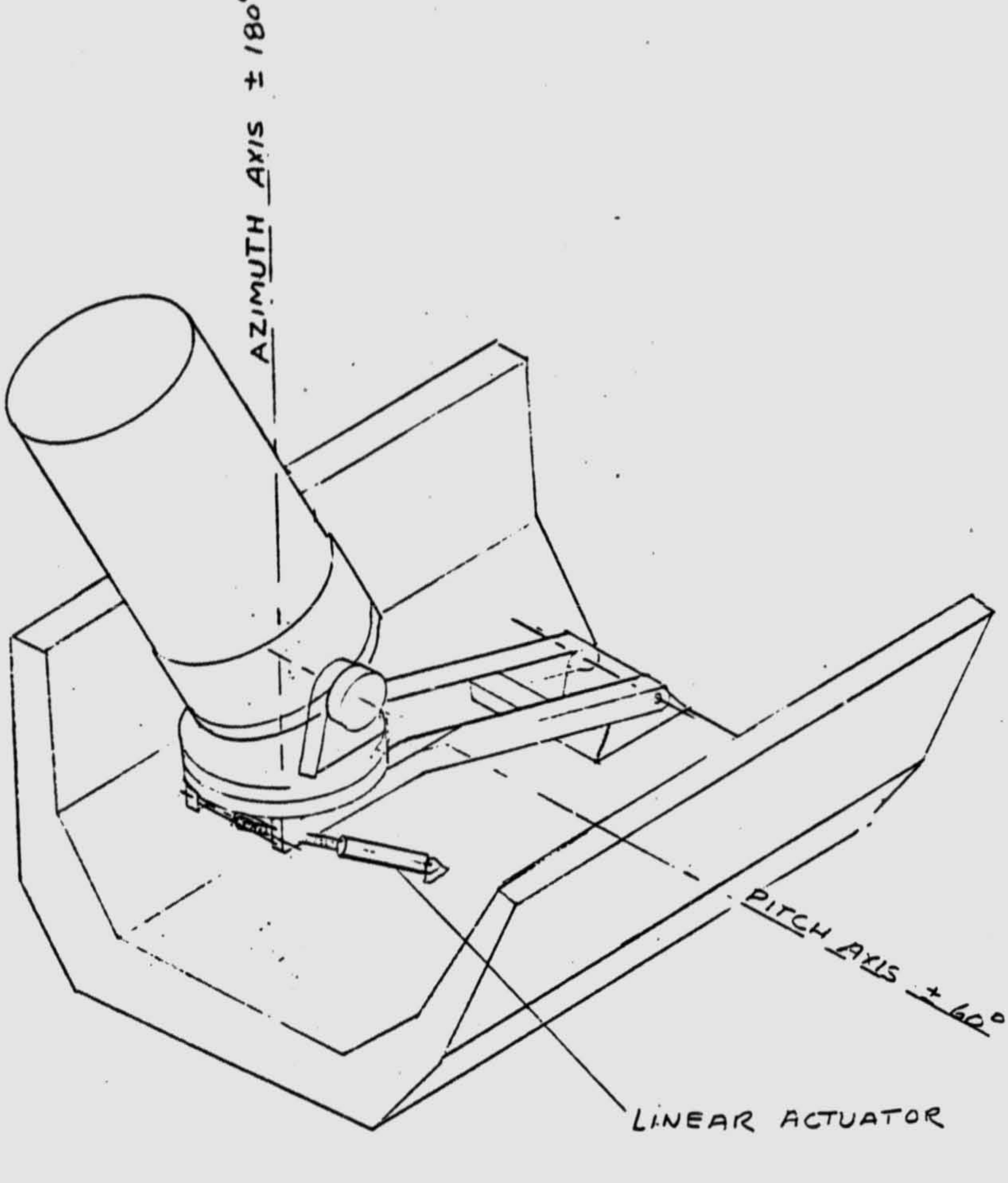

FIGURE 3-4 CONCEPT # 4 (END MOUNT, DEPLOYABLE)

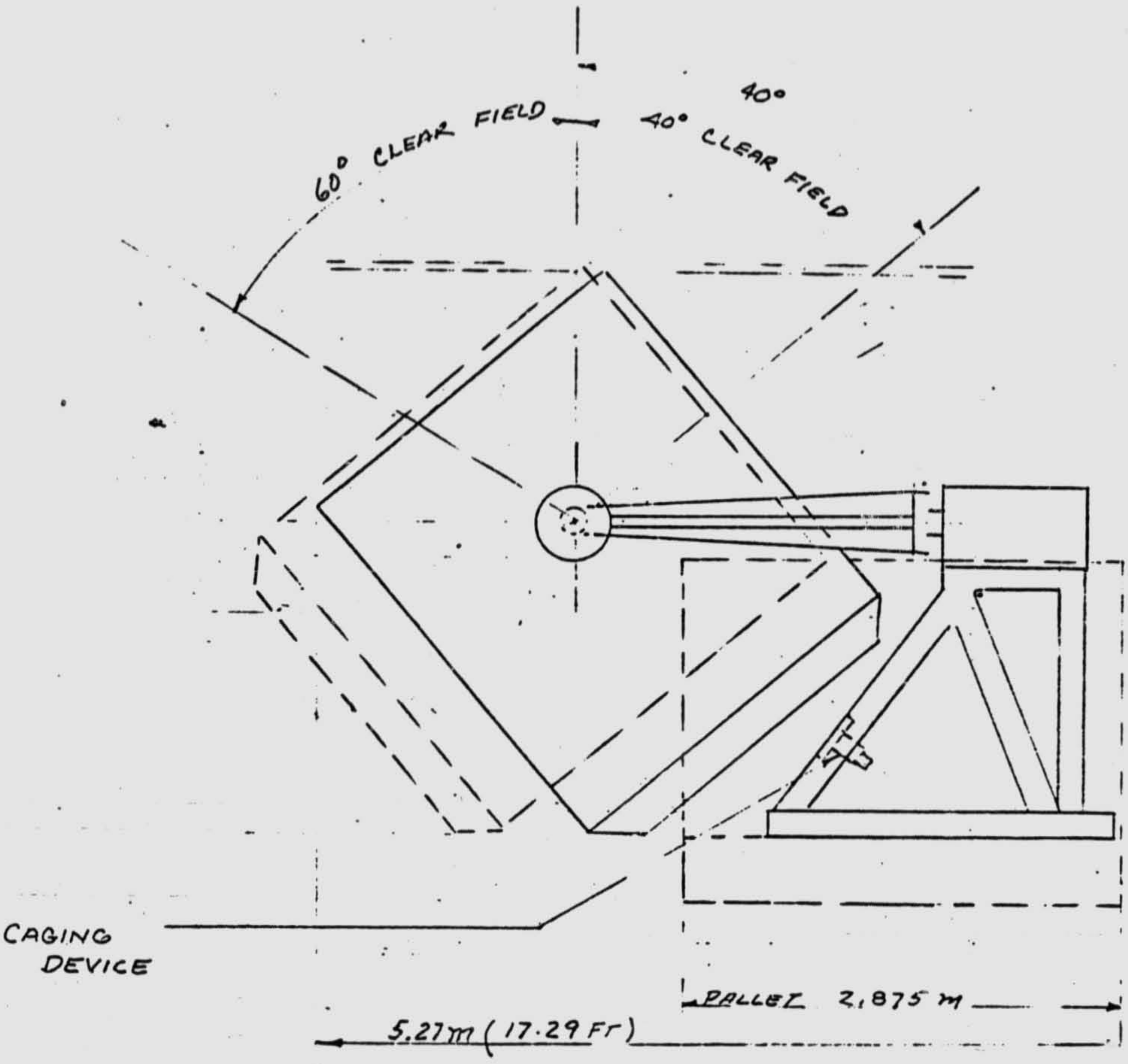

FIGURE 3-5 CONCEPT # 2A (ROLL-PITCH, FOR LARGE PAYLOADS)

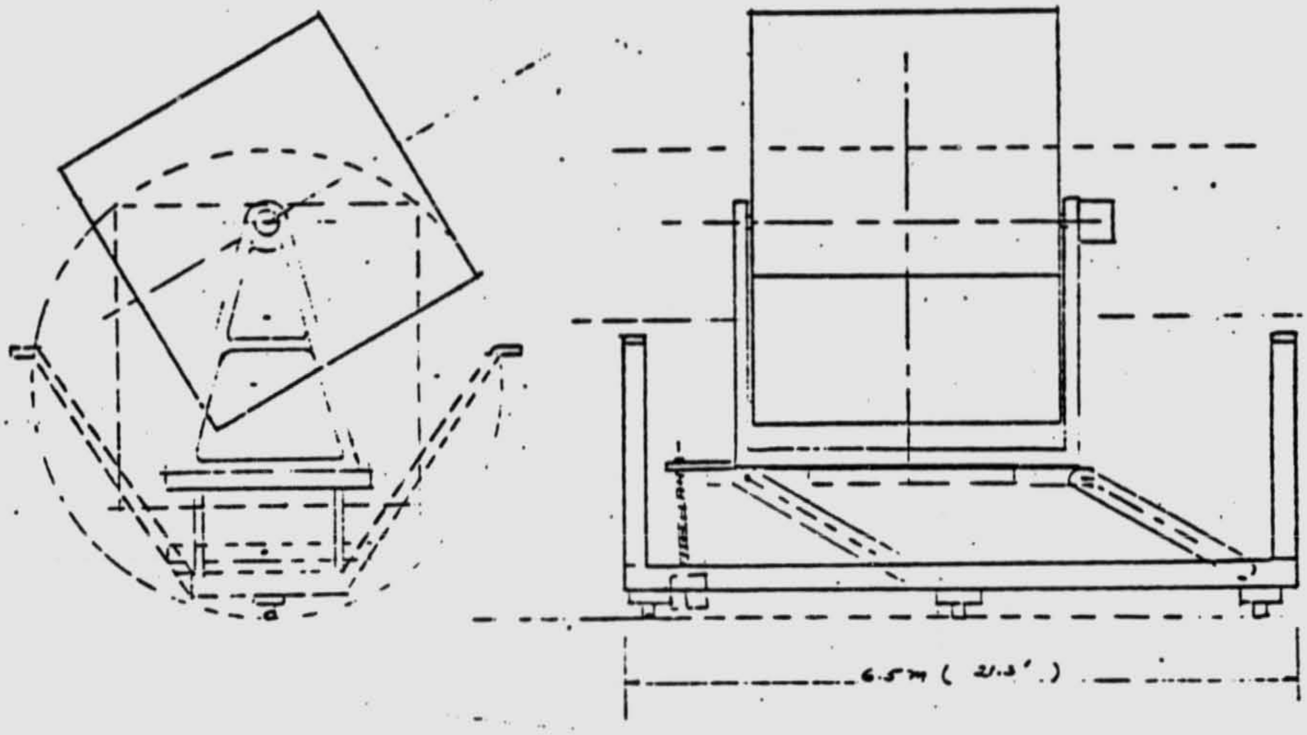

FIGURE 3-6 CONCEPT # 4A (AZIMUTH - ELEVATION, ORBITER MOUNTED)

CONCEPT #2 (Roll-Pitch)

This is also a c.g. mount, but pointing control is more complex for a roll-pitch configuration. Depending on instrument size, a significant overhang over the pallet will result for the full 60° FOV requirement. Pallet loading is non-symmetrical.

CONCEPT #3 (End Mount)

This configuration has the potential for a large FOV for possible instruments, but raises the overall height of the FIM/instrument package. This could require jettison capability if the Orbiter cargo bay envelope is penetrated during on-orbit operations. This configuration is no a c.g. mount but picks up instrument loads at an instrument end plate.

CONCEPT #4 (End Mount, Deployed)

For larger payloads it is required to deploy the end mount. This adds significant complexity to the system and requires jettison capability. Again, this is not a c.g. mount.

CONCEPT 2A (Roll-Pitch,

This configuration is the same as Concept #2, except that it is configured for a large instrument. The result is even larger pallet overhang, the length requirement in the Orbiter cargo bay would be equivalent to almost 2 pallet lengths.

CONCEPT 4A (Azimuth - Elevation Axes, CG Mount, Non-Pallet)

This concept uses its own support structure to tie directly into the Orbiter cargo bay trunnion and keel fittings. It is intended for large payloads and requires ~~deployment beyond the cargo bay trunnion and keel fittings. It is intended for~~ large payloads and requires deployment beyond the cargo bay envelope for full FOV. The length required in the cargo bay is almost 2 pallet length. Provisions have to be made for accommodation of Spacelab subsystem equipment (e.g. RAU's, EPDB) normally located on the pallet.

It should be noted, however, that a support structure located on the pallet other than the spacelab pallet might be an attractive alternative again if the availability of pallets is severely constraint.

3.2 ACCOMMODATION OF LARGE INSTRUMENTS

Large instruments with maximum dimensions of 2m x 2m x 3m length, and 2m diameter and 3m length present a definite problem as illustrated in Figure 3-7. Shown is a 2m diameter, 3m long instrument on the pallet within the cargo bay envelope. This requires not only launch and landing of the instrument in a special stowed position, but also an in-orbit deployment mechanisms to achieve the required FOV, and emergency jettison capability in case a deployed instrument cannot be retracted before Orbiter re-entry and landing.

3.3 FEASIBILITY CONSIDERATIONS

In view of the requirements and design guidelines/groundrules discussed in Section 2, it is clear that concepts #4 and 4A present a significant step in overall complexity and potential development cost compared to the non-deployable, pallet mounted concepts. They were, therefore, not considered any further.

Concept 2A, because of its large pallet overhang, potentially complex pallet interface and requirements for complex payload retentions during launch and landings, was also eliminated from further consideration.

Concepts 1, 2 and 3 were further evaluated, as described in Section 4.

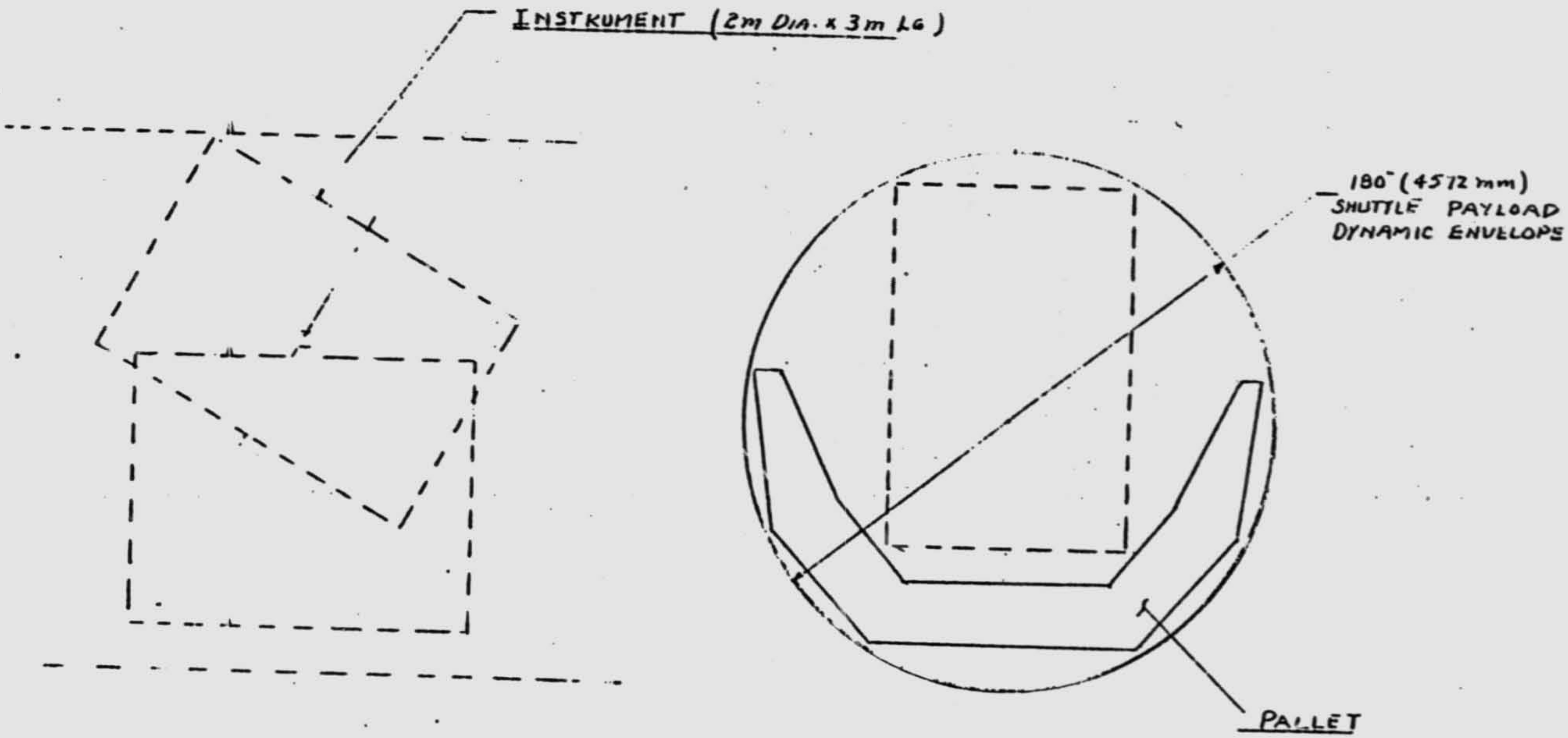

Figure 3-7 Comparion (Instrument/Payload Bay)

4. CONCEPT EVALUATION (TASK 3)

The preferred concept was selected by evaluating Concept #1, 2 and 3 against evaluation criteria which were derived from the requirements identified in Section 2.

The following were determined as the most important evaluation criteria (not in order or priority)

1. Accommodation of payload sizes as identified in Section 2, i.e. 24 LAMAR modules and cylindrical payloads of 2m diameter and 3m length.
2. Minimum required length in the Orbiter cargo bay, i.e. maximum utilization of the Spacelab pallet volume and mounting area.
3. Staying within the Orbiter cargo bay payload envelope during all on-orbit operational phases to avoid emergency jettisson requirements.
4. Full 60° field of view.
5. Provide instrument interface to an instrument central support ring.
6. Ground testing without extensive need for GSE.
7. Simple and standard interfaces limited to Spacelab.

It was already pointed out in Section 3 that the 3m length requirement is very difficult to meet without violating the cargo bay payload envelope, for any FIM configuration using the Spacelab pallet. This is, of course, the result of the basic cargo bay shape and of the fact that the Spacelab pallet itself occupies a significant amount of the useable cargo bay volume.

It was clear, therefore, that a compromise had to be accepted as far as instrument length was concerned.

<u>CONCEPT #3</u> (End Mount)

This concept is best suited for instruments which can interface with the FIM at an instrument end plate. For instruments requiring support at an instrument ring

(close to the instrument c.g.) additional support structure would be necessary. In any case, a large c.g. off-set is inherent in this concept.

While Concept #3 could potentially meet evaluation criteria 1,2,3, 4 and 7 as well as the other two concepts, it clearly does not meet criteria 5 and 6. Since ease of integration, ground testing and minimum requirements for GSE are essential for a low cost system, Concept #3 was not considered for a more detailed engineering definition.

<u>CONCEPT #2</u> (Roll-Pitch)

This concept provides a c.g. mount and supports the instrument at a central support ring.

In order to provide the required FOV for an instrument of about 2 m dia. and the max. possible length within the cargo bay payload envelope, a significant pallet overhang in the order of 1.5 m would be required. Depending on the particular Spacelab configuration, this would either result in a significant infringement of the payload envelope on a neighboring pallet (e.g. in a 2 or 3 pallet train), or in the requirement to locate a neighboring pallet sufficiently separated to avoid the interference.

Obviously, this is not an efficient utilization of the Orbiter cargo bay space.

Another disadvantage of the roll-pitch concept is the highly unsymmetric load introduction into the roll-bearing in 1-g conditions during ground testing. Since Concept #2 doesn't offer any clear advantages over Concept #1, the azimuth-elevation mount, in any of the other evaluation criteria it was also not considered for a subsequent engineering definition.

CONCEPT #1 (Azimuth - Elevation)

Measured against the evaluation criteria established above, the azimuth-elevation drive concept meets all criteria and was established as the clearly preferred concept. It was subsequently defined in more detail in Task 4, Engineering Definition, which is described in the following section.

In detail, the evaluation criteria are met as follows:

1. Instrument Size - The FIM described in the following section (see Fig. 5-1 to 5-6) can accommodate cylindrical payloads of up to 2.25 m dia. (incl. a thermal cannister) and 2.5 m length. In the central part of this cylinder (about 1.6 m dia.) the instrument length can be extended to about 2.8 m. This envelope is sufficient to accommodate 24 LAMAR modules, and easily accommodates the other two gamma-ray instruments identified for this study.

2. Minimum Cargo Bay Length - The FIM/instrument assembly, for the full 60° FOV, does not extend outside of the pallet payload envelope.

3. Orbiter Cargo Bay Payload Envelope - The FIM/instrument assembly stays within this envelope during all operational phases.

   Larger payloads which would penetrate this envelope require that the FIM/ instrument assembly can be jettissoned.

4. 60° FOV - The full 60° FOV as defined in Section 3 is achieved.

5. Instrument Interface - The selected FIM concept provides a support ring which interfaces with an instrument central support flange as required, near the ~~instrument~~ c.g.

6. Ground Testing - Is possible without extensive need for supporting GSE because of the completely symmetrical configuration and loading of the large azimuth bearing.

7. STS Interfaces - The preferred FIM concept uses the pallet hardpoints, i.e., the standard pallet structure interface, for pallet mounting. No direct tie-ins to the Orbiter are required. In case a detailed load analysis shows that a latching mechanism is necessary to support the FIM instrument launch/landing loads, this latching mechanism can either be part of the FIM support structure or might at most use some additional pallet hardpoints (see Section 5).

5. ENGINEERING DEFINITION (TASK 4)

5.1 INTRODUCTION

Having selected the preferred configuration of a symmetrical elevation/azimuth gimbal arrangement, the mechanization of the concept was then further investigated. Specific electromechanical components were selected on a preliminary basis and performance predictions established in respect to the requirements specified in Section 2. This section describes these components with some rationale for their selection.

5.2 PRELIMINARY SYSTEM DESIGN

The system design is primarily based upon the basic design requirements discussed in Section 2, and summarized in Table 2-3.

Since weight was not considered critical the system is designed to be rugged, capable of withstanding launch and landing loads without damage to the equipment. The structural analysis of the Flexible Instrument Mount is shown in Appendix A.

The system consists of a structural subsystem to support the experiment with a standardized attachment for the experiment, an electromechanical drive subsystem, and an electronic controller for instrument offset pointing.

MECHANICAL DESIGN

The primary objectives of the design were to (a) establish the maximum payload size that can be accommodated, with the selected FIM concept approach, (b) develop a feasible concept for supporting the structure incldthe interfaces and (c) suggest further study areas for more definitive level to assess the concept credibility.

PAYLOAD SIZE

Determination of the payload size was done in the following steps:

1. Outline the shuttle payload envelope using a standard pallet.
2. Examine if the payload mentioned in the RFP (2M x 2M x 3M) can be accommodated with a complete 60° cone rotation and remaining within the shuttle envelope.
3. If the above fails, construct a payload size which will fit.
4. Develop a support structure concept for the payload with the required drive mechanisms included.
5. Examine the feasibility of the concept.

The concept finally evolved is shown in the Applicon drawings. (Fig. 5-1 to 5-6) The payload size of 2.25 M diameter and 2.5 M length can accommodate 24 modules of the LAMAR experiment and also the other two Gamma Ray experiments without exceeding the cargo bay dynamic envelope.

THE FIM STRUCTURE

The FIM Structure consists of: (See Figure 5-1)

1) Payload interface ring
2) Yoke - side arms
3) Bottom Ring structure - bearing interface
4) Bearing housing
5) Pallet interface
6) Caging device

The pallet interface ring is a circular ring with a rectangular hollow section. Thin ring with a rectangular hollow section. A thin ring provides the payload attachment points along its circumference. It may also provide hardpoints for the caging device pick up during the launch and landing modes (See Figure 5-5). This ring has two rigidly attached steel shafts at diametrically

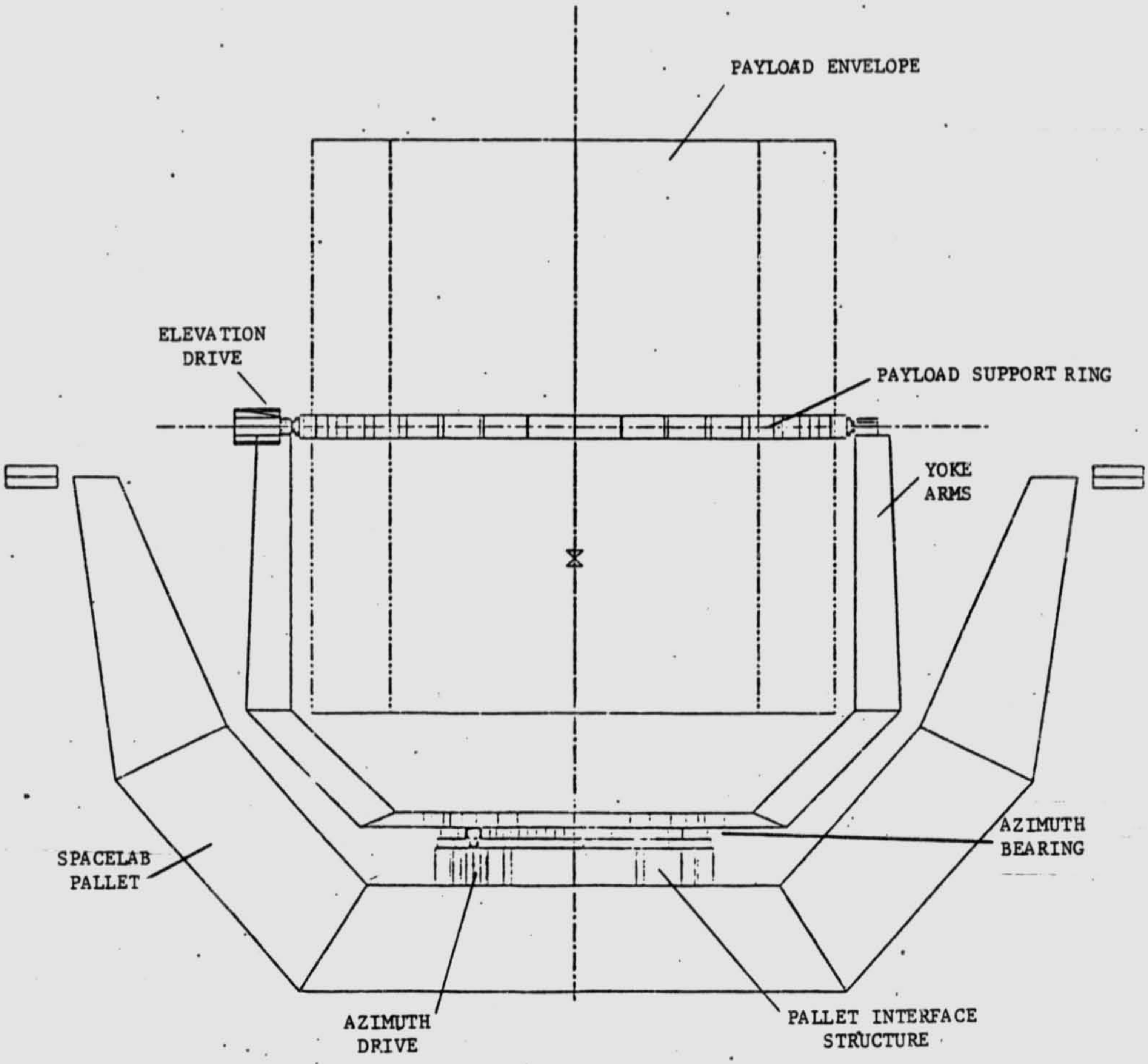

FIGURE 5-1 FIM STRUCTURE

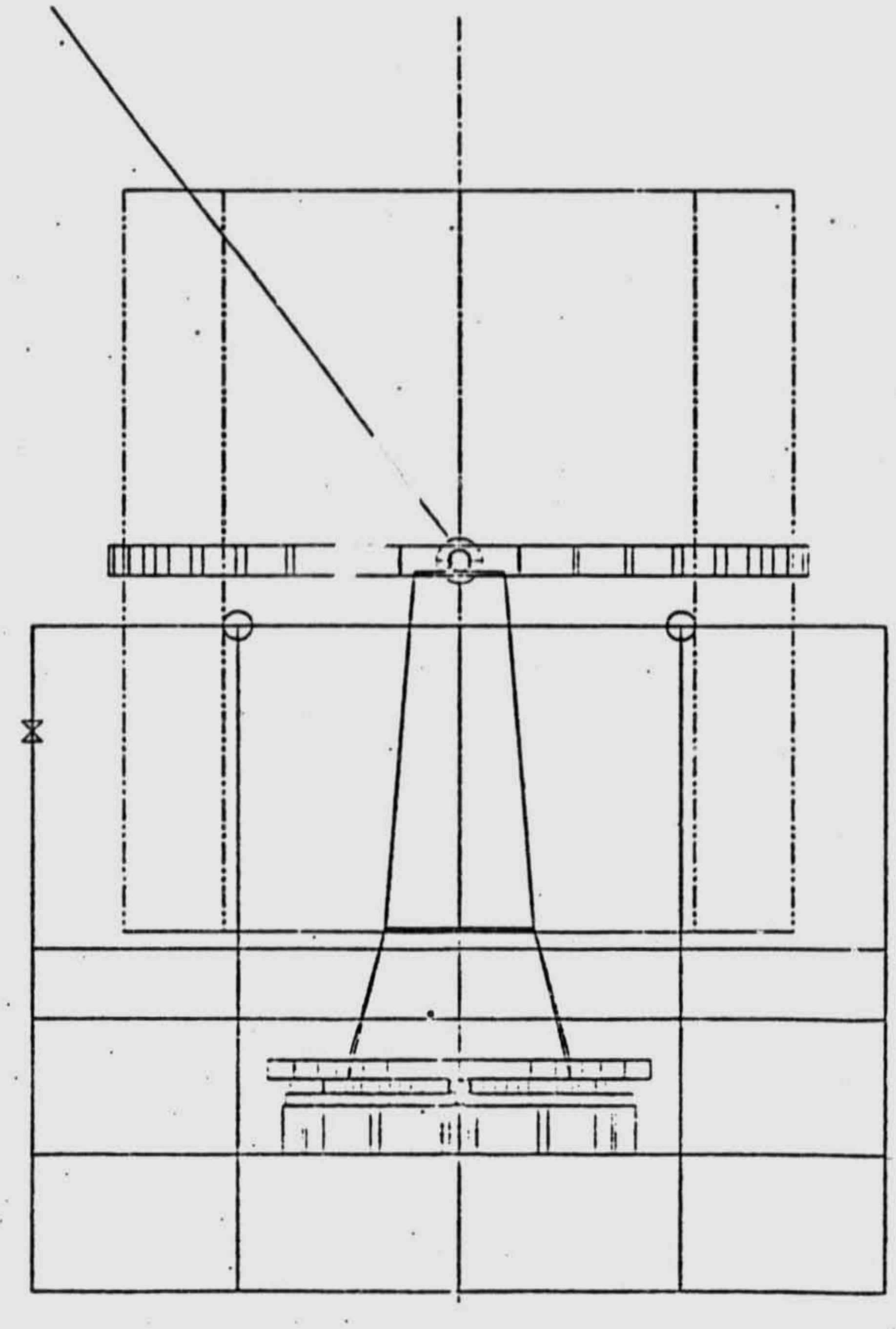

FIGURE 5-2

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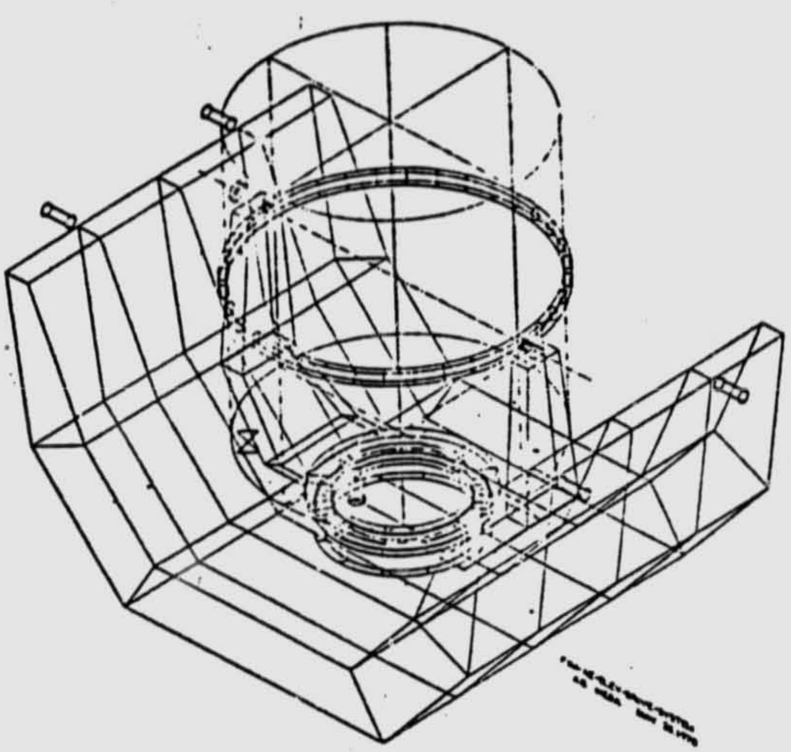

FIGURE 5-3

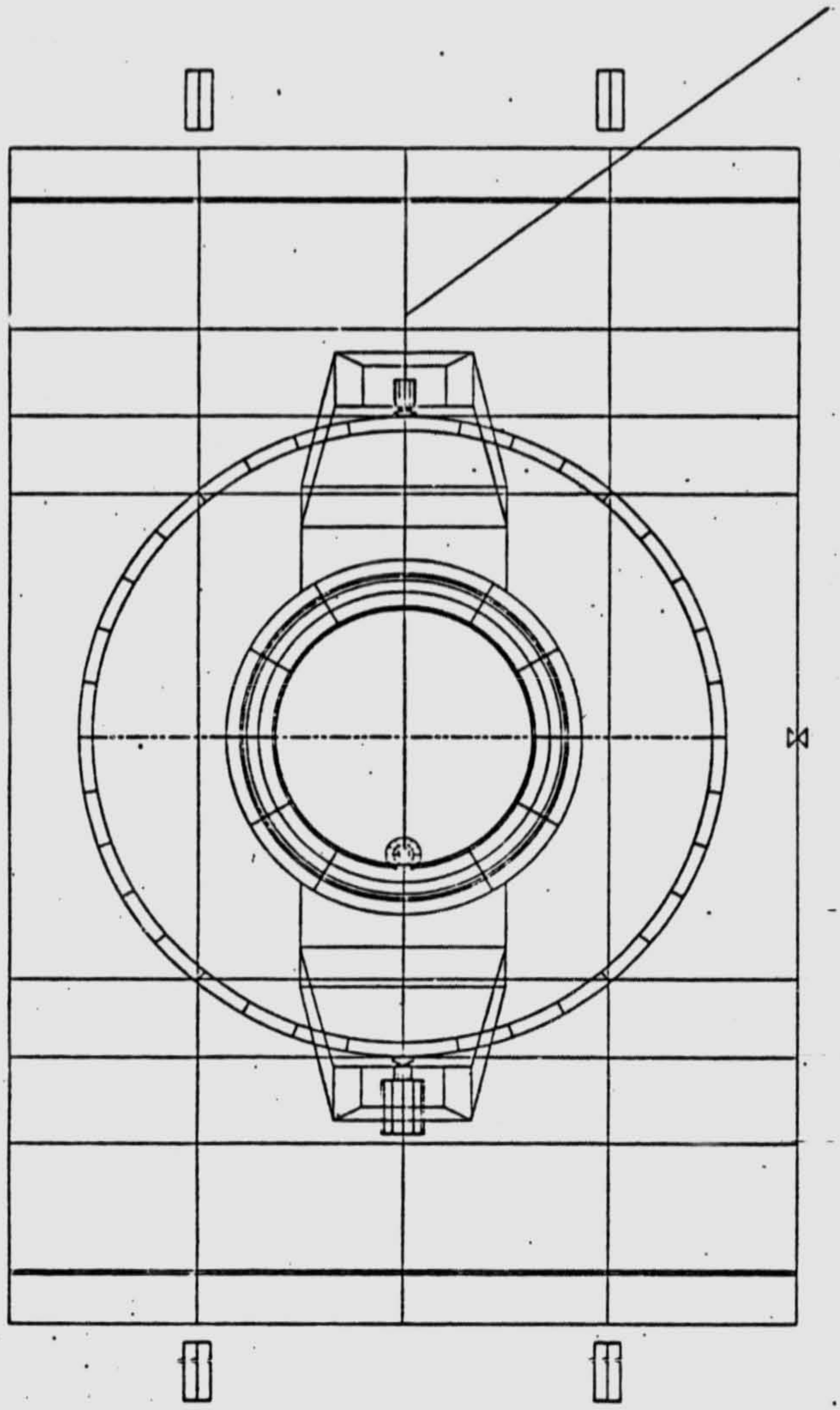

FIGURE 5-4

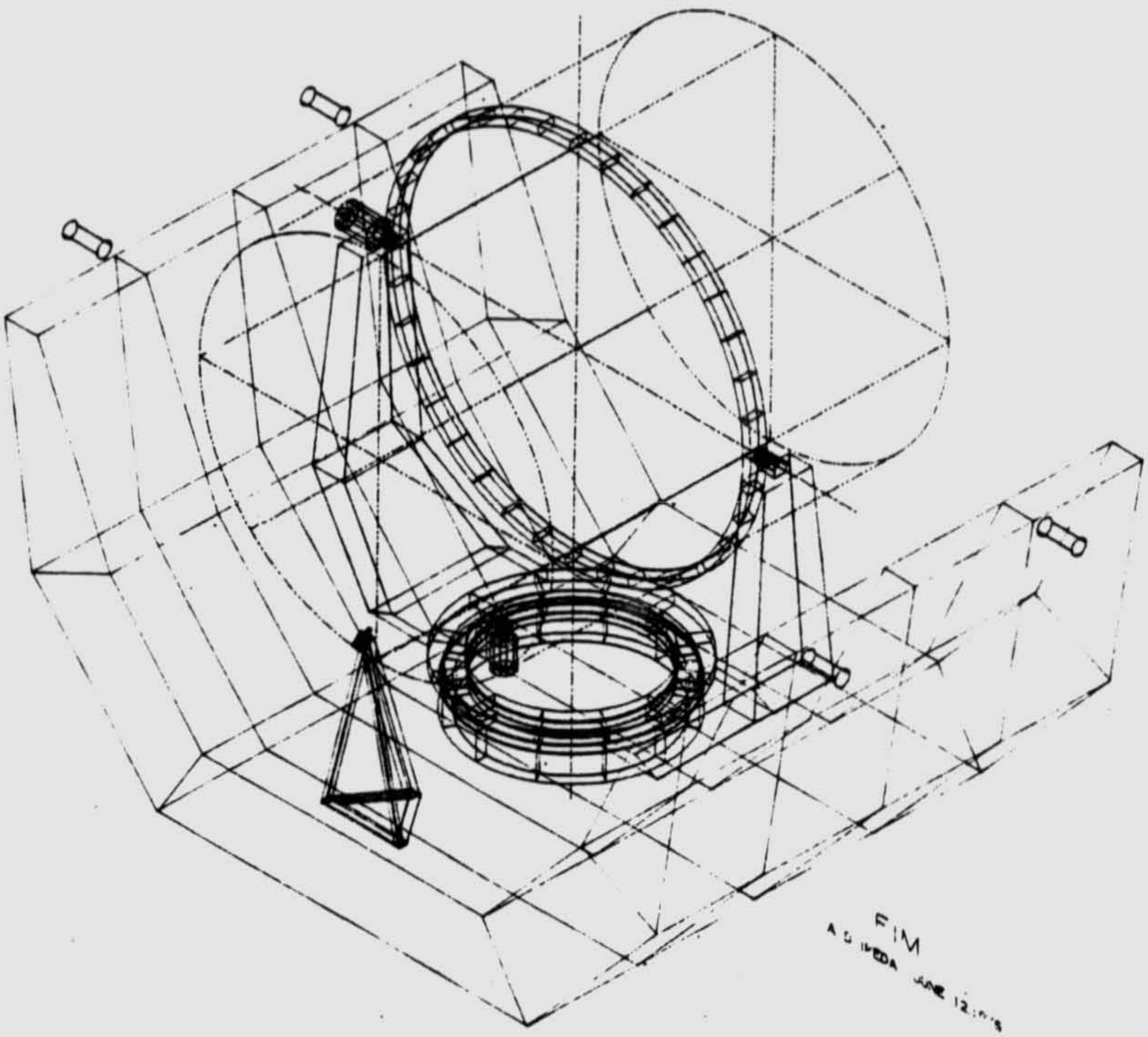

FIGURE 5-5
(ISOMETRIC, INSTRUMENT IN HORIZONTAL POSITION

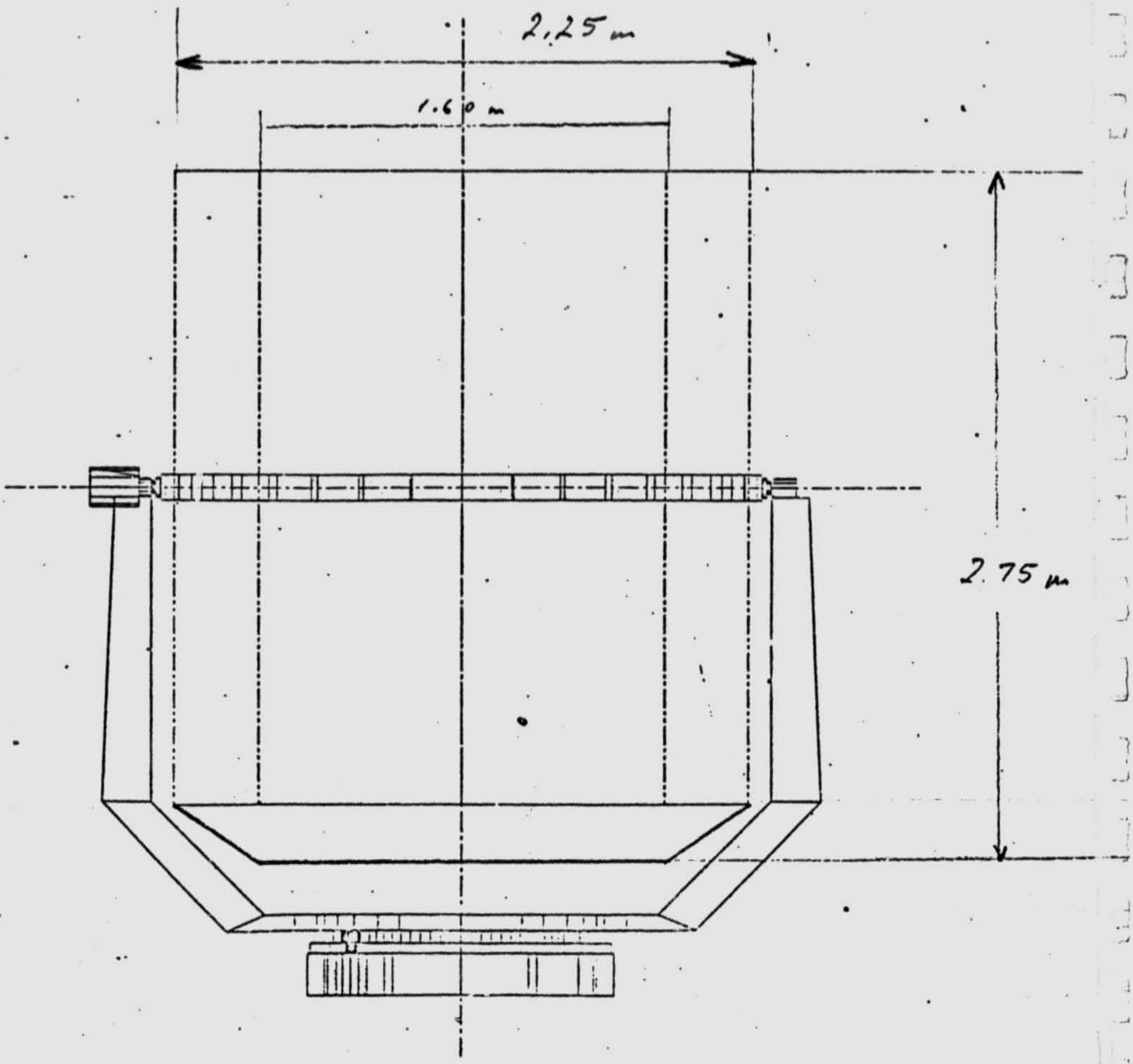

FIGURE 5-6

opposite locations, which are (the shafts) supported by bearings mounted on the Yoke arms.

THE YOKE ARMS

The Yoke arms form the main body of the FIM structure. As shown in the drawings each member supports the payload ring at one end and the other interfaces with the bottom ring structure supported on the azimuth bearing. The cross section of any arm is hollow rectangular of varying outer dimensions, narrower at the payload ring connection and wider at the bottom ring connection.

Preliminary structural analysis (See Appendix A) of the Yoke and the payload ring indicated the necessity of additional load support for structural stiffness, during the launch and landing modes. However no significant stress problem was encountered.

The additional structural support can be provided through the caging device discussed in the following pages.

BOTTOM RING STRUCTURE

The bottom ring structure forms the seat of the whole payload. It is also in the form of a ring mounted on to the azimuth bearing. This bearing interface needs further study.

FIM-PALLET INTERFACE CONCEPT

A preliminary FIM-PALLET interface concept is shown in Figure 5-7. The primary mounting link consists of a rigid metal frame made of commercially available steel or aluminum profiles.

The FIM azimuth bearing is rigidly mounted onto this frame, through its bearing housing. This assembly of the bearing and frame is then interfaced with the pallet load bearing hardpoints.

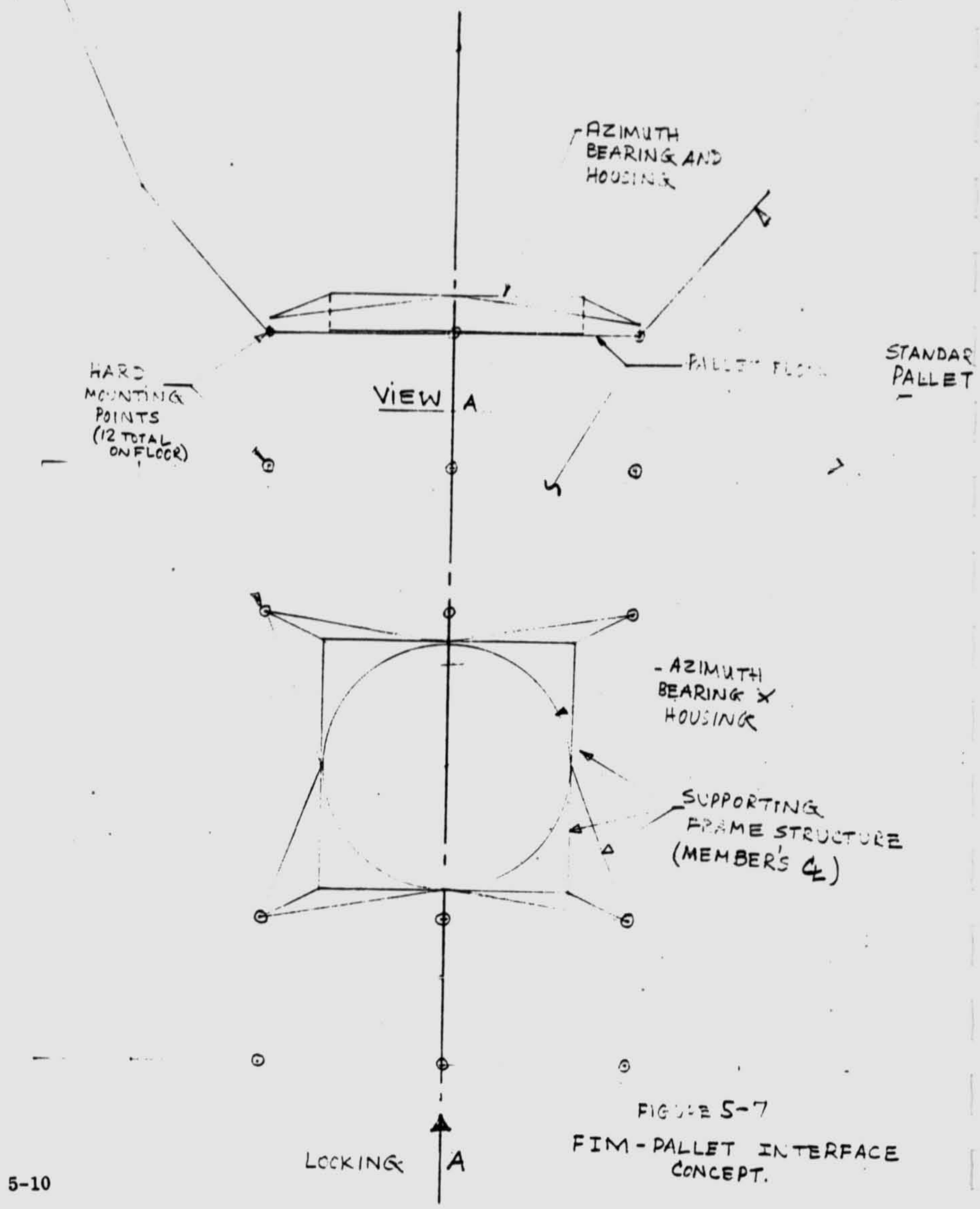

FIGURE 5-7
FIM-PALLET INTERFACE CONCEPT.

The baseline concept would be to use the six keel and inner longeron hardpoints on pallet frame 2 and 3 (See Figure 5-8). If payloads exceeding the capability of these six hardpoints have to be accommodated, additional floor hardpoints on frame 1 and 4 can be used by modular extensions of the interface frame.

If final analysis shows that floor hardpoints are not sufficient, then the use of outer longeron or sill hardpoints has to be considered. A preliminary hardpoint load calculation is included in Appendix A.

It might be possible again to use a modular extension of the basic interface frame to go to these hardpoints. This, however, should be traded-off against a concept that employs the foreseen caging mechanism to transfer loads preferably into the sill hardpoints which are most effective for load transfer into the Orbiter attach fittings.

The problem with determining how many hardpoints are needed to transfer payload loads into the pallet - e.g. for the maximum FIM configuration - is the lack of adequate pallet load carrying capability data in SPAH.

The values given in SPAH refer only to the ultimate load capability of the primary pallet structure at each particular hardpoint location. In other words, the load that any given hardpoint sees in an acutal case is the load introduced into that hardpoint by the payload, plus loads distributed by the pallet structure into that hardpoint - due to loading of other hardpoints on the pallet. What this means, for cases when the payload mass approaches the "nominal" pallet load carrying capability, is that a coupled pallet/payload structure analysis has to be carried out. This requires, of course, the availability of at least a simple finite element model of the Spacelab pallet which was not available.

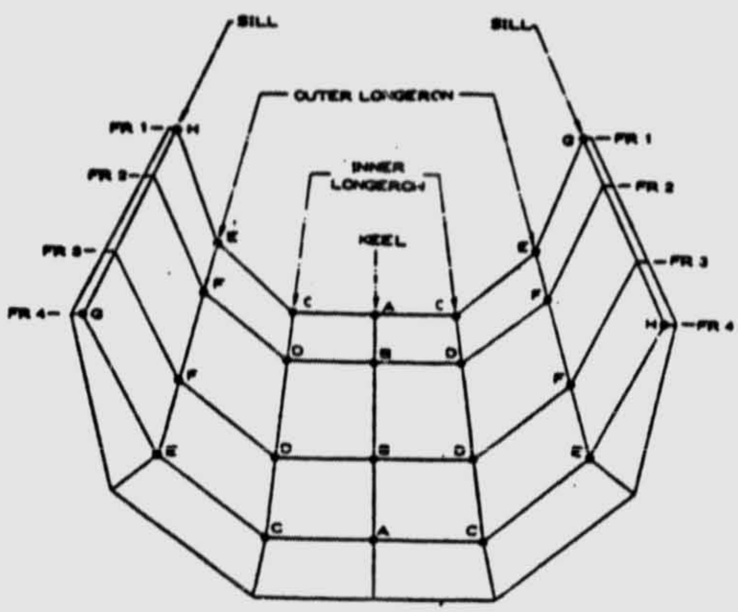

FIGURE 5-8 HARDPOINT ATTACHMENT LOCATIONS (FROM SPAH)

Based on the ultimate load carrying capabilities of each hardpoint as stated in SPAH, and on pallet/payload coupled analysis for a very heavy payload carried out by ERNO/HSD some time ago, we are confident that our proposed simple interface structure with modular extension capabilities will be capable of transferring FIM/payload shown in Appendix A loads into the pallet.

To accommodate the deflections of the pallet during thermal and mechanical environments, the hardpoint ball shank provides for an articulation within a 14 degree cone of arc. However, further study is required to determine more closely the actual relative deflection problems, its implications and to optimize the hardpoint integration capability.

CAGING DEVICE

As mentioned earlier caging devices are required during the launch and landing modes. These devices have mainly two purposes: 1) Increase the stiffness of the primary structure, and 2) provide load paths to additional hardpoints..

REQUIRED FURTHER STUDY

1. Caging Device
2. Pallet interface
3. Azimuth Bearing Housing on support
4. Bottom ring structure interface. i.e., transfer of load from the yoke arms to the azimuth bearing.

5.3 PRELIMINARY FIM COMPONENT SELECTION

The components of the system have been selected to meet the system requirements with simplicity and low cost in mind. Some of the component characteristics are summarized in Table 5-1.

TABLE 5-1

PRELIMINARY

DEFINITION OF COMPONENTS

<u>DRIVE UNIT</u>

Type	DC Torquer with "Orbidrive" Reducer
Size	5" dia. x 6" long
Weight	10 lbs.
Torque	
Operating	158 ft. lbs. Capacity @ 120°/min
Stall	240 ft. lbs.
Gear Ratio	1400:1 (Direct Drive) 135:1 (Pinion Drive)
Motor	Inland T-2967 (or equiv.)

<u>BEARINGS</u>

Azimuth	Rotek Model	L9-42N1Z (or equiv.)
	O.D.	47.20"
	I.D.	36.20"
	WT.	435 lbs.
	Thrust Cap.	248,000 lbs.
Pitch	Fafnir	5214 (or equivalent
	O.D.	4.92"
	I.D.	2.75"
	WT.	4.62 lbs.
	Radial Cap.	21,400 lbs.

<u>POTENTIOMETER</u> Helipot Multiturn - D

O.D. 3.00"

~~WT 17~~ oz./Linearity ~~.025%~~

Bearings - The gimbal bearings are an important factor in the mechanization task. If the azimuth bearing is sized to withstand a worst case loading of 5g by itself without damage a very large bearing is required. The equivalent thrust load rating required is derived from the equation;

$$\text{Freq.} = F_T + \frac{4.37\ M}{D} + 3.44\ F_R$$

where

F_T = actual thrust load

M = moment load

F_r = Radial load

D = Raceway Dia

Feq = equivalent thrust load

(See Appendix B for ROTEK Bearing Calculation Material)

Assuming a load of 4400 lbs. (see load geometry shown belos) with an acceleration of 5g acting at an angle of 45° we have

$$F_T = F_R = 4400 \times 5 \times .707 = 15{,}500$$

If load 6.6 offset is 6.23 ft.

$$M = 15{,}500 \times 6.23 = 96{,}900 \text{ ft. lbs.}$$

$$\text{Feq.} = 15{,}500 + \frac{4.37 \times 96{,}900}{3.5} + 3.44 \times 15{,}500$$

$$\text{Feq.} = 190{,}028 \text{ lbs.}$$

This load is in excess of rated load even on a 4.75 ft. diameter Rotek bearing. However, a reasonable compromise is to size the bearing for normal loads of 5g. This results in a landing load of approximately 86,167 kg (190,000 lbs.) of equivalent thrust. A Rotek bearing L9-42 (3.5' raceway dia.) has a thrust rating of 248,000 lbs. providing a safety factor of 1.3. This analysis assumes that the bearing is taking full load. Retention devices can normally be designed to absorb some load thus increasing the margin of safety.

Therefore the Rotek L9-42 bearing appears to be a satisfactory preliminary choice for the azimuth axis and would be compatible with the maximum weight experiment. It is possible that this bearing would be damaged by 9g crash loads and would need to be replaced. This bearing is illustrated in Figure 5-9.

The bearing on the elevation (pitch) axis will not experience any overturning moment and need only to have a radial capacity of 20,000 lbs. The Fafnir bearing part no. 5214 as described in Table 6.2 would be a good candidate for this function.

DRIVE UNIT

A drive unit incorporating a torquer motor and a form of an harmonic speed reducer has been tentatively selected for a drive unit for both the azimuth and elevation axes. This device is shown conceptually in Figure 5-10 the Orbidrive reducer manufactured by compudrive corp., North Billerica, Mass. employs eccentric to generate harmonic motion. However, the rotary motion is derived from rugged sprocket cam and rolled technique as contrasted to the meshing of fine teeth found in other drives. High torques are reached by increasing cam size and high reduction ratios are obtained by a compact multistage construction. The operating principle of this drive is described in Figure 5-11. The drive is tentatively selected for FIM applications because of its high torque capability and relative simplicity. A short development may be required to fully adapt this device to aerospace usage but low unit cost is the ultimate benefit thereafter. GE has received a layout and budgetary quotations from Compudrive on this drive unit for FIM.

The torquer motor was chosen because of its high torque and high slew speed capability. Stepper motors were considered for the drive but do not have adequate output torque at the 120°/min slew rate requirement particularly at ground test conditions. Directly connected torquers would provide a simple

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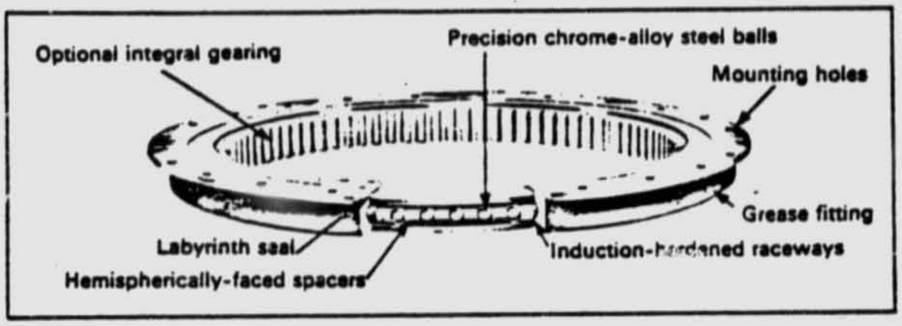

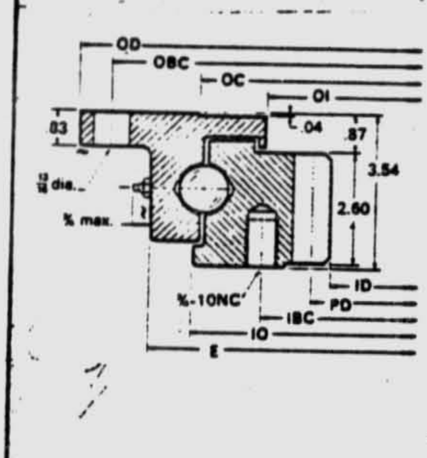

Figure 2

FIGURE 5-9 PROPOSED AZIMUTH BEARING (ROTEK INC.)

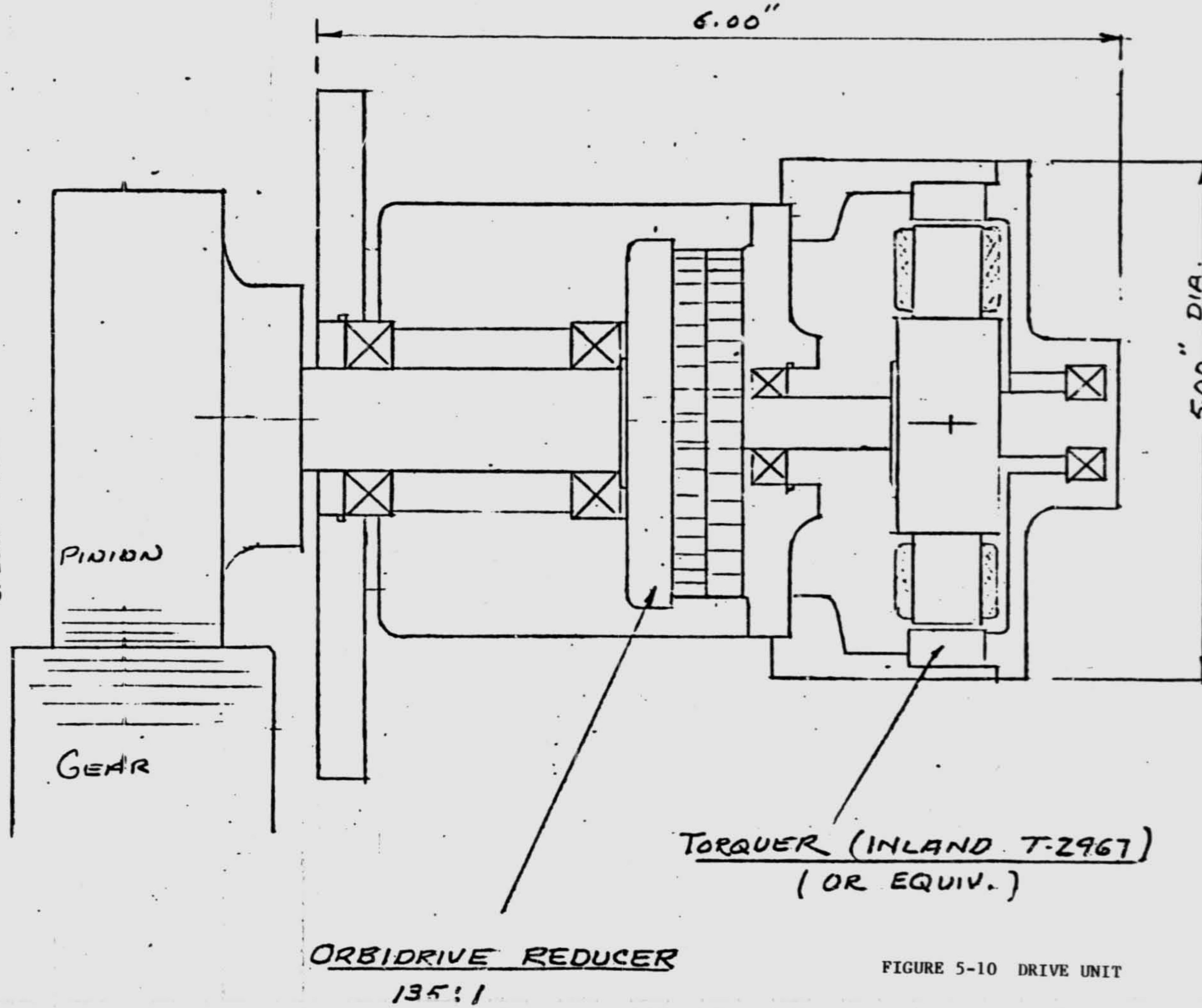

FIGURE 5-10 DRIVE UNIT

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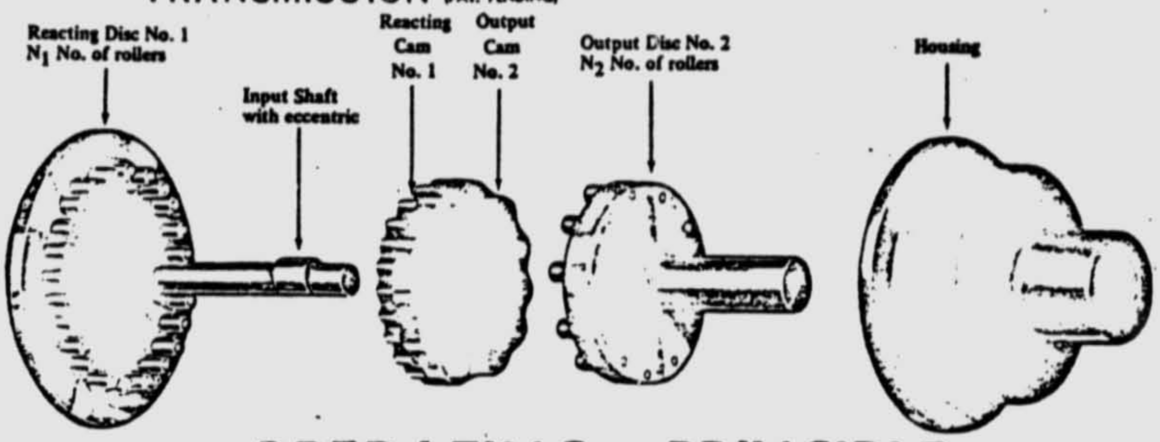
THE
Orbidrive®
TRANSMISSION (PAT. PENDING)

A dramatic new low cost, high efficiency gearless drive for speed reducers.
Shown below: essential components of the Orbidrive transmission; bearings omitted for simplicity.



OPERATING PRINCIPLE

Operation of the Orbidrive transmission is based upon the rotational relationship of two interlocked multi-lobed cam and roller sets, each having a different number of lobes and rollers. They are so linked that one cam/roller set drives the other at a differential velocity proportional to the number of cam lobes and roller contacts in each set.

The two cams (see diagram) are fastened together concentrically and mounted on the eccentric shaft. Input torque applied to the eccentric shaft causes cam number 1 to orbit within disc number 1 which is rigidly affixed to the housing. Each revolution of the eccentric advances cam number 1, one cam lobe position on disc number 1. Since cam number 2 is fastened to cam number 1, it too is advanced the same number of rotational degrees. With continued rotation, lobes of cam number 2 orbit against rollers of disc number 2, imparting a rotation to the disc which is proportional to the ratio of rollers in disc number 1 to disc number 2. This can be calculated from the formula:

$$\frac{(N_1 - 1)N_2}{N_1 - N_2}$$

in which N_1 is the number of rollers in one cam/disc set and N_2 is the number of rollers in the other cam/disc set.

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FIGURE 5-11 ORBIDRIVE TRANSMISSION (OPERATING PRINCIPLE)

approach but the size and weight required to handle the azimuth load of approximately 480 in lbs. at 1g, (i.e., 60 lbs. for motor above with peak power of 800 watts) is obviously prohibitive.

The gear ratio of 1400: 1 for the azimuth axis was selected by optimization of output torque at slew rate of 120^{o}/min. The output torque of the drive versus motor speed and total gear ratio is plotted in Figure 5-12 shown by the dotted curve. Note that this curve peaks at an effective gear ratio of 1400: 1 providing a torque of 160 ft. lbs. a good margin over even a wide range of load torquer from the azimuth bearing at ground test conditions.

POSITION FEEDBACK

An encoder readout of 0.1^{o} is required for position control of the instrument. If an analog sensor is used, a multiturn pot such as the Helipot Multiturn D series unit would be required having a precision linearity of 0.025% since 0.1^{o} is .027% of 360^{o}. The pot would have to be gear driven with step-up gear ratio. Since gearing will add additional errors, a digital encoder should be considered as an alternate. For instance an Itek Digisec, size 35 with 2^{12} resolutions (5.3 min. of arc) is a possible choice. This unit is 3.5" in diameter and is available in a solid shaft or hollow shaft configuration. It is recommended that final choice of the position sensing device be integrated with the servo contract design.

5.4 FIM CHARACTERISTICS

WEIGHT

Based on the selected FIM structural concept, siting of the structured components (See Appendix A); and selection of the main FIM components total system weight was estimated as shown in Table 5-2.

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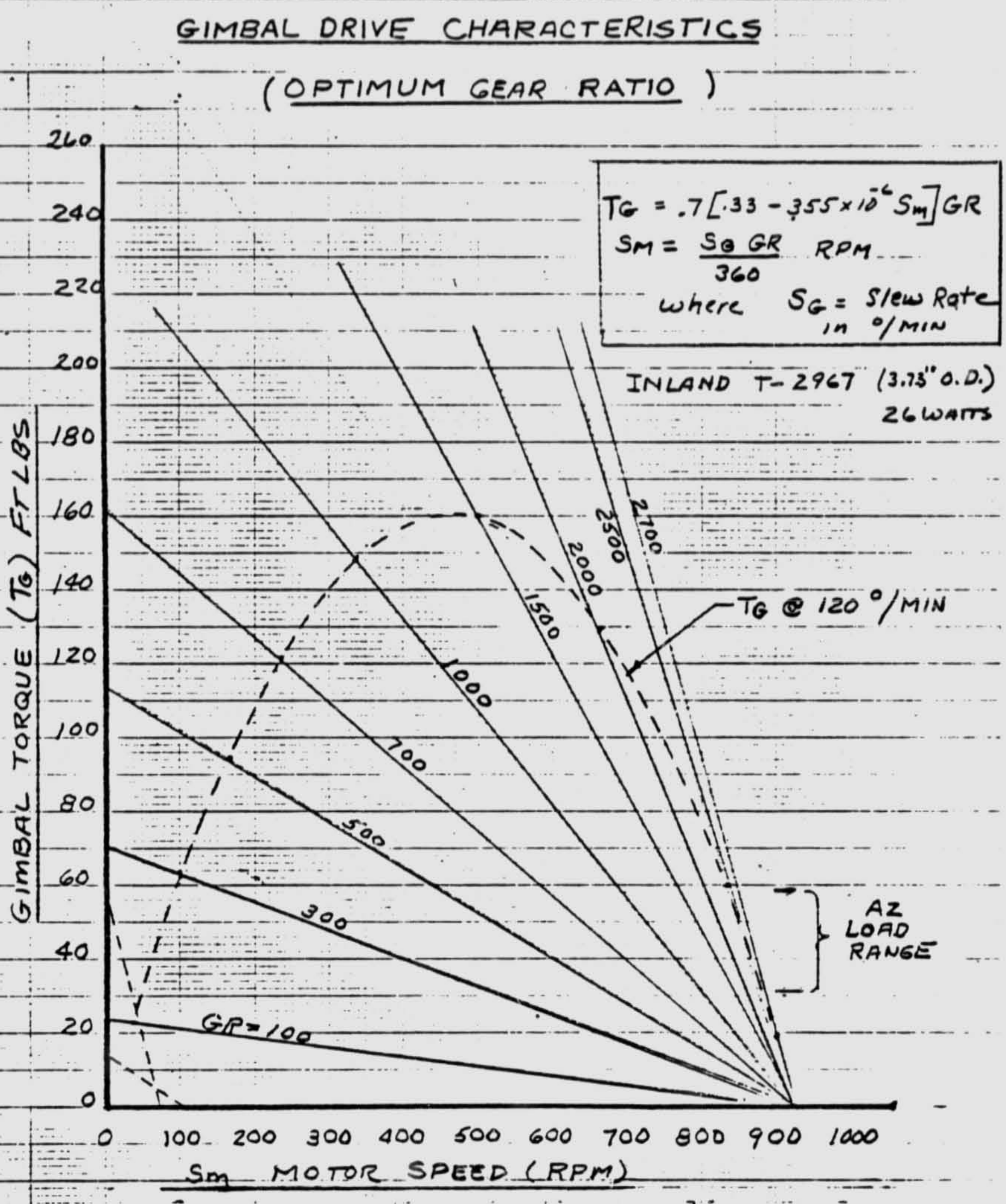
FIGURE 5-12



POWER

The FIM electrical power requirements are estimated to be:

Maximum @ Motor Stall	=	70 Watts
Holding Power in Orbit	=	3 Watts
Slew Power	=	35 Watts

EXPERIMENT CG OFFSET

The drive torque characteristics of the drive system operating in a 1g field largely determine the allowable amount of offset of the center of gravity of the instrument. The following limitations apply to the proposed concept for worst case conditions assuming that Zero-G simulations are not to be used:

- o Offset critical only for Ground Operations
- o Relatively Large Offsets are allowable in orbit, but will require secure caging for launch and landing
- o C.G. offset of 2.0" Max. allowed for Ground Test without Zero-G aids. (i.e., Static balance within 4,000 in. lb. for 2,000 lb. experiment.

If larger offsets are present in heavy experiments GSE support equipment would be required for ground test.

TABLE 5-2 FIM WEIGHT ESTIMATES

STRUCTURE		1320	
COMPONENTS		470	
THERMAL COVER		30	
CAGING DEVICE		30	
MISCELLANEOUS		20	
	TOTAL	1870	(850 Kg)
PAYLOAD (ASSUME)		3300	(1500 Kg)
	TOTAL	5170	(2350 Kg)
MAXIMUM PALLET CAPACITY		6600	(3000 Kg)
	GROWTH	1430	(650 Kg)

SERVO ANALYSIS AND ERROR BUDGET

Based on a very conceptual control system design and preliminary servo analysis, as discussed in Appendix C, the following in-orbit error budget was established (tentative).

ERROR BUDGET (In Orbit)

	AZ	EL
Electronics (Servo)	0.50°	$.027^\circ$
Gear Backlash	$.03^\circ$	$.036^\circ$
Potentiometer Drive	$.02^\circ$	$.02^\circ$
Experiment Alignment	$.10^\circ$	$.10^\circ$
SUM	.65	.183
RSS	.51	.11

5.5 SAFETY ASPECTS

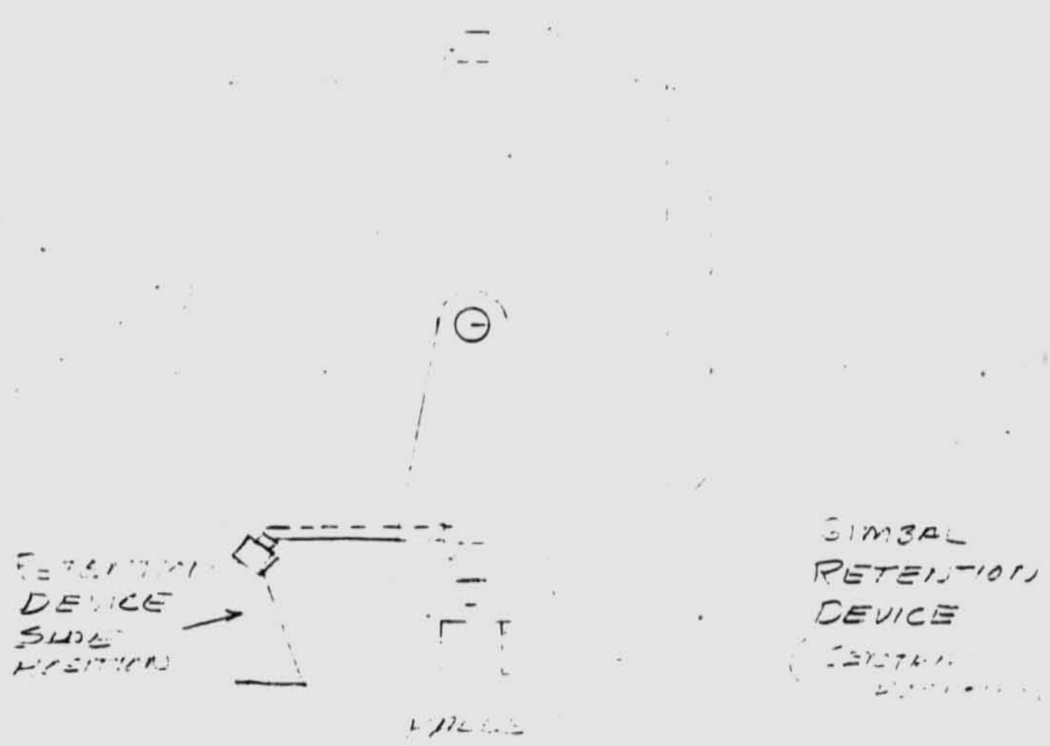
There are three important safety requirements that are important for FIM.

1. Maintain structural integrity during emergency landings of the orbiter, i.e., stay attached to the Spacelab pallet and do not disintegrate.
2. Provide emergency jettison capability in case any part of the FIM/payload system can penetrate the orbiter cargo bay envelope during on-orbit operations.
3. Retain FIM from moving during launch and nominal landing operations, i.e., allow safe landing was in case FIM cannot be commanded back into a landing configuration offer on-orbit operations.

The emergency jettison capability is not required since the FIM/payload envelope is restricted to stay within the orbiter cargo bay envelope at all times. In order to retain FIM from movement during launch and landing maneuvers, it appears desirable to restrain the pitch gimbal with a device mounted to the pallet so that it may share some of the load imposed on the azimuth. One concept is shown below with the retention device located on

the azimuth axis. A side position is also shown as an alternate. Further study is needed to determine the method of retention which will provide the best support for the experiment along with greatest safety for equipment and personnel.

As discussed in paragraph 5.3, the bearing would be capable of withstanding the crash load without structural failure; however, distortions would take place that would require replacement of the bearing after a crash landing. Thus the launch lock could fail and still have the assembly stay in tack.



6. FIM DEVELOPMENT PLAN (TASK 5)

The final study task generates a development plan and estimates the resources required to design, fabricate, test, and maintain one protoflight model of the FIM. In addition, a suggested series of "Phase B" studies is identified to provide detailed design data for the hardware (Phase C/D) procurement activity. A bottoms-up cost estimate is presented as a separate submittal.

The first step is to develop a detailed Work Breakdown Structure for the FIM Phase C/D effort. This WBS is shown in Figure 6-1. The three left-hand elements (Program Management, System Engineering, and Product Assurance) contain tasks that pertain to all hardware programs; their specific organization can vary from program to program and is a matter of preference. For example, "Operability" under System Engineering in this WBS refers to concerns such as maintainability, usability, transportability, and on-orbit performance, while in other WBS's it might also include reliability and safety. The important thing is to be certain that all Level 3 elements are included for schedule and cost considerations.

The central element of the WBS (Subsystem Engineering) is most specific to FIM and for this reason it is broken out to Level 4 for each subsystem. The Engineering Definition from Task 4 provides the technical basis for this breakout.

The next two elements (GSE and System Test) are significant cost and schedule drivers and are also broken out to Level 4. The final element (Operations Support) assumes that FIM integration and operations are carried out by someone other than the FIM developer, so that the developer's involvement in operations is limited to logistics, maintenance, and sustaining engineering. These activities are broken out as Level 3 elements.

A FIM hardware development schedule is presented in Figure 6-2. This schedule allows 18 months for design, manufacturing, and test of the FIM protoflight model. Operations in support of experiment to FIM integration requires another 9 months. Maintenance, refurbishment, and storage in support of a second flight will be scheduled to conform with the flight dates; normal FIM maintenance and refurbishment is expected to require no more than two weeks.

Prior to Phase C/D hardware development, it is desirable to perform a Phase B preliminary design. A Phase B WBS is given Figure 6-3. It includes all the Level 2 elements of the Phase C/D WBS as well as the Level 3 Subsystem Engineering elements. Further Level 3/Level 4 breakouts include design and planning elements based on the Phase C/D WBS.

A "modular" Phase B activity is presented that addresses the Phase B WBS in a series of related studies in order to fit into a definition program with funding constraints. Those WBS elements that bear most heavily on the Phase C/D procurement package are planned for earlier studies. Elements that "fine tune" the design or address secondary concerns are delayed until later studies. These later studies could proceed in parallel with the Phase C/D procurement activity so long as their results are available at the Phase C/D Preliminary Requirements Review.

A candidate FIM development schedule containing modular Phase B studies is shown in Figure 6-4. This schedule assumes a first FIM flight in mid-FY 1983 and the initiation of Phase C/D procurement activities in mid-FY 1980. Brief descriptions of each potential Phase B study are given below:

System Capability Design looks at designing the FIM to meet specific performance requirements. The study includes:

- System Requirements/Design
  - Pallet Interface Design

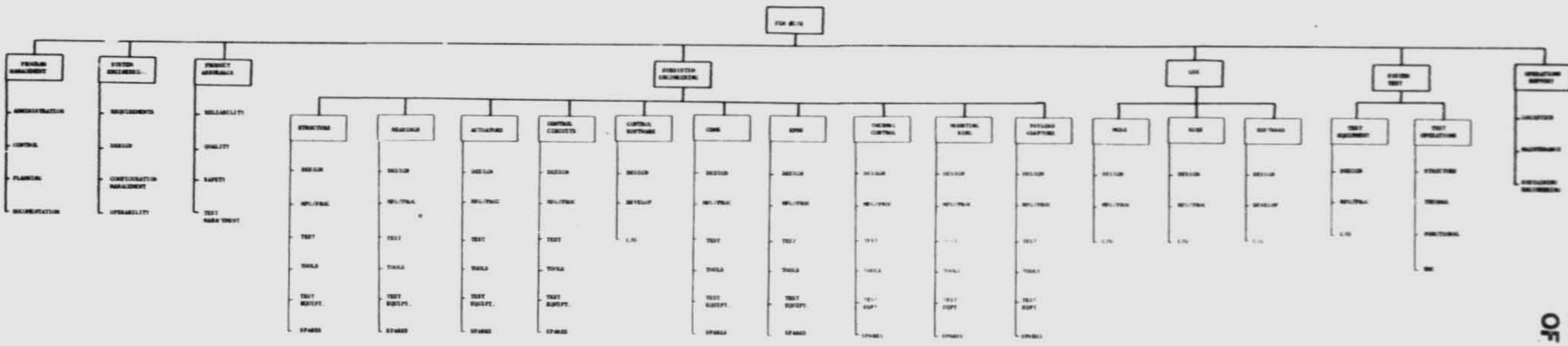


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FIGURE 6-1 WBS PHASE C/D

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YEAR 1	YEAR 2	YEAR 3
	TBD	

FIGURE 6-2 FIM HARDWARE DEVELOPMENT SCHEDULE

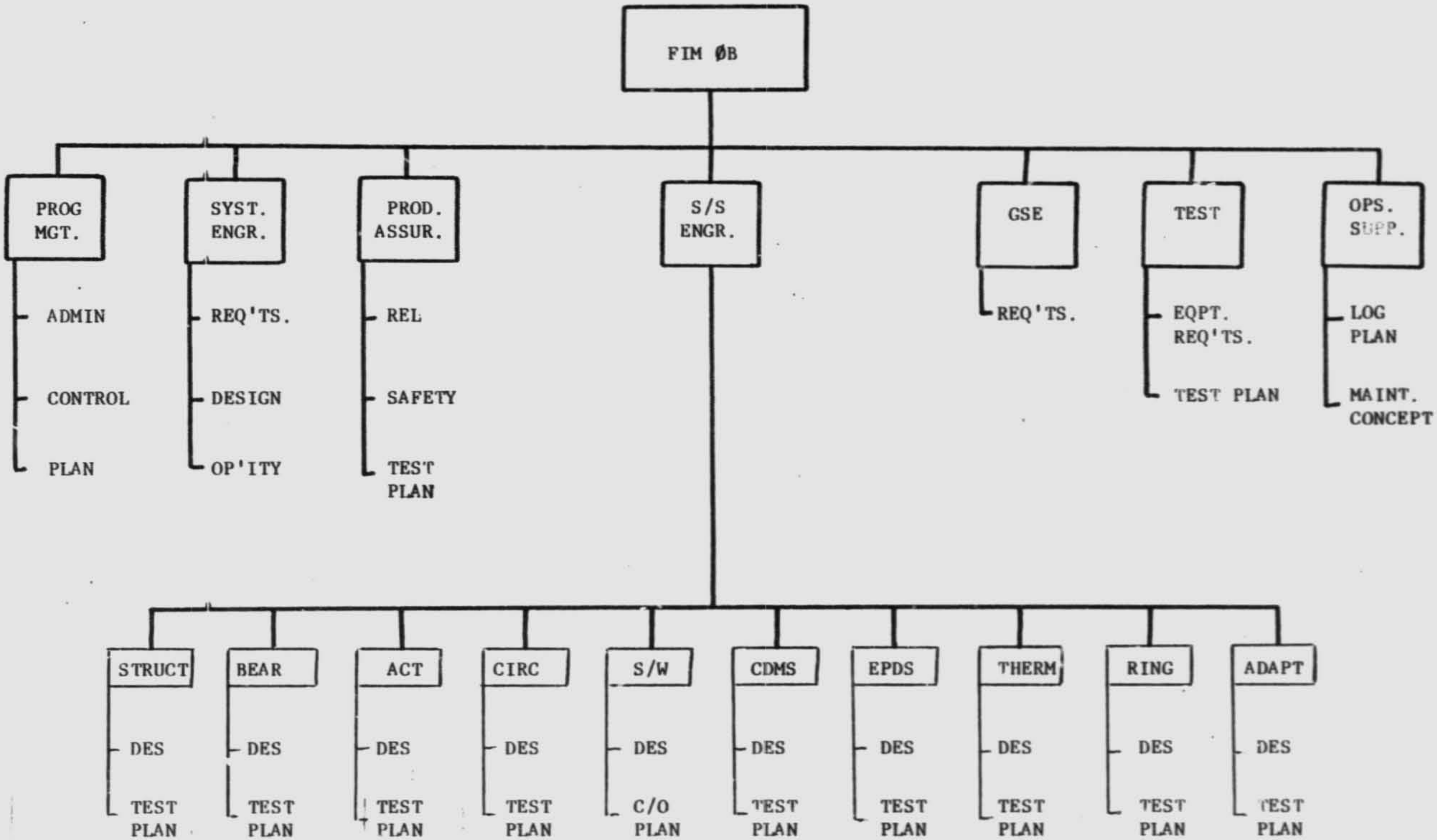


FIGURE 6-3 WBS PHASE B

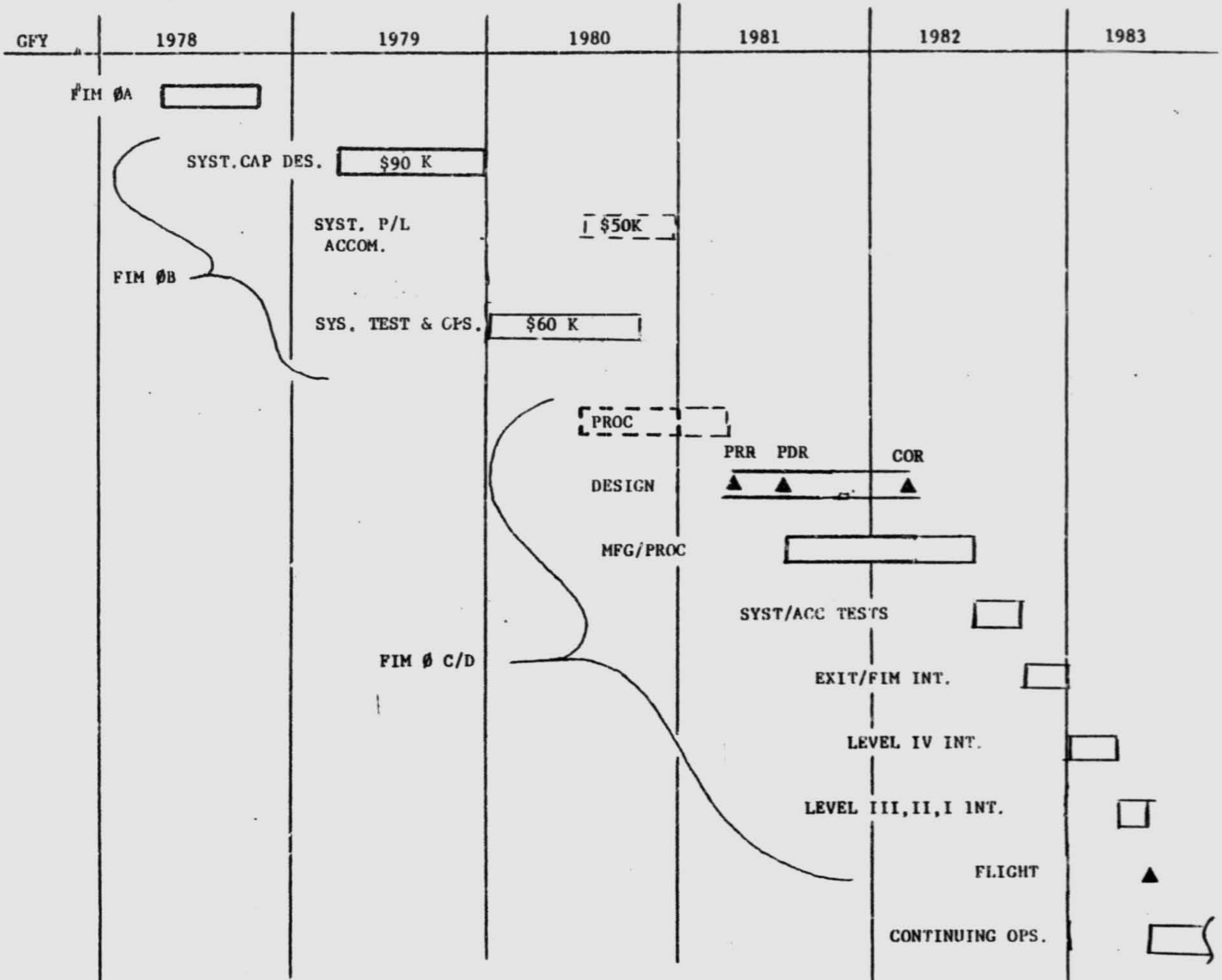


FIGURE 6-4 FIM DEVELOPMENT SCHEDULE

- Structure Design and Analysis
- Bearing Design
- Actuator Design
- Control Circuitry Design
- Control Software Design
  - Dynamic Simulation
- FIM Thermal Analysis
- System Test and Checkout Concept
- System Operation and Maintenance Concept
  - Operability
- System Reliability and Safety Requirements
- Program Management Plan
- GSE Requirements

System Test and Operations looks at the details of test, checkout, operations, and maintenance for the Phase B FIM design. This activity can proceed in parallel with Phase C/D procurement activities. The study includes:

- System Test Management Plan
- Component Test Requirements
- Subsystem Test Requirements
- System Test Requirements
- System Operability
  - Number of Flight Units
- GSE Requirements
- Logistics Plan
- Maintenance Concept

System Payload Accommodation looks at adapting the FIM design to a range of possible experiments. It may be undertaken jointly with one or more experiment programs or it might possibly be done entirely by the experiment

programs. Results should be available for the Phase C/D PRR. This study should include:

- System Requirements/Design
  - Experiment Interface Design
- Mounting Ring Design
- Payload Thermal Control
  - Thermal Can
- CDMS Interface Design
- EPDS Interface Design

APPENDIX A

FIM STRUCTURAL ANALYSIS

INTRODUCTION

THE FOLLOWING STRUCTURAL ANALYSIS OF THE FLEXIBLE INSTRUMENT MOUNT WAS PERFORMED USING THE "FEAST" (FINITE ELEMENT ANALYSIS OF STRUCTURES) COMPUTER PROGRAM. "FEAST" IS A GENERAL DESK-SIDE COMPUTER PROGRAM FOR THE LINEAR ELASTIC STATIC ANALYSIS OF COMPLEX THREE DIMENSIONAL STRUCTURES.

A FINITE ELEMENT MODEL OF THE FLEXIBLE INSTRUMENT MOUNT WAS CONSTRUCTED FOR USE IN "FEAST". IT CONSISTS OF 10 NODES, 6 STRAIGHT BEAMS AND 4 CURVED BEAMS. THE WEIGHT OF THE STRUCTURE WAS DISTRIBUTED OVER ALL ELEMENTS OF THE MODEL AND THE 4400 LB. PAYLOAD WEIGHT WAS DISTRIBUTED OVER THE FOUR CURVED BEAMS WHICH COMPRISE THE RING OF THE FLEXIBLE INSTRUMENT MOUNT.

A 1 g LOADING IN THE Z-DIRECTION (OUT OF PLANE OF STRUCTURE) WAS APPLIED TO THIS "FEAST" MODEL. TO PERFORM A STRESS ANALYSIS, THIS 1g OUT OF PLANE LOADING WAS MULTIPLIED BY AN ULTIMITATE LOAD FACTOR OF 9.0 FOR SHUTTLE EMERGENCY LANDING. THE RESULTING STRESSES WHERE THEN COMPARED TO ULTIMATE ALLOWABLES. THE RESULTS OF THIS COMPARISON ARE SHOWN IN TABLE I AND THE ALLOWABLES ARE SHOWN IN TABLE II. TO CHECK THE

NATURAL FREQUENCY OF THE STRUCTURE, THE DEFLECTIONS FROM 1g LOADING WERE USED TO CALCULATE AN APROXIMATE NATURAL FREQUENCY OF THE STRUCTURE AS A CANTELEVER BEAM. THIS FREQUENCY WAS THEN COMPAIRED TO A REQUIRED MINIMUM NATURAL FREQUENCY OF 5.0 * HERTZ. THIS CALCULATION IS SHOWN ON P. 8 + 9.

FROM THE MARGINS OF SAFETY SHOWN IN TABLE I, IT IS OBVIOUS THAT THE STRESSES FOR THE STRUCTURE ARE ACCEPTABLE FOR THE CONDITION CONSIDERED. THE NATURAL FREQUENCY OF 10.9 HERTZ CALCULATED ON P. 9 IS ALSO ACCEPTABLE.

* It is desirable to keep the natural frequency of pallet mounted instruments above 25 Hz. For lower frequencies a payload/pallet coupled analysis has to be performed. This analysis has not taken the caging mechanism into account (see Section 5.2) which would increase the natural frequency reported here.

TABLE I - MARGINS OF SAFETY

LOCATION	BENDING	SHEAR + TORSION
RING	+7.65	AMPLE
SHAFT	+0.20	NOT CRITICAL
UPPER YOKE	AMPLE	AMPLE
LOWER YOKE	AMPLE	AMPLE

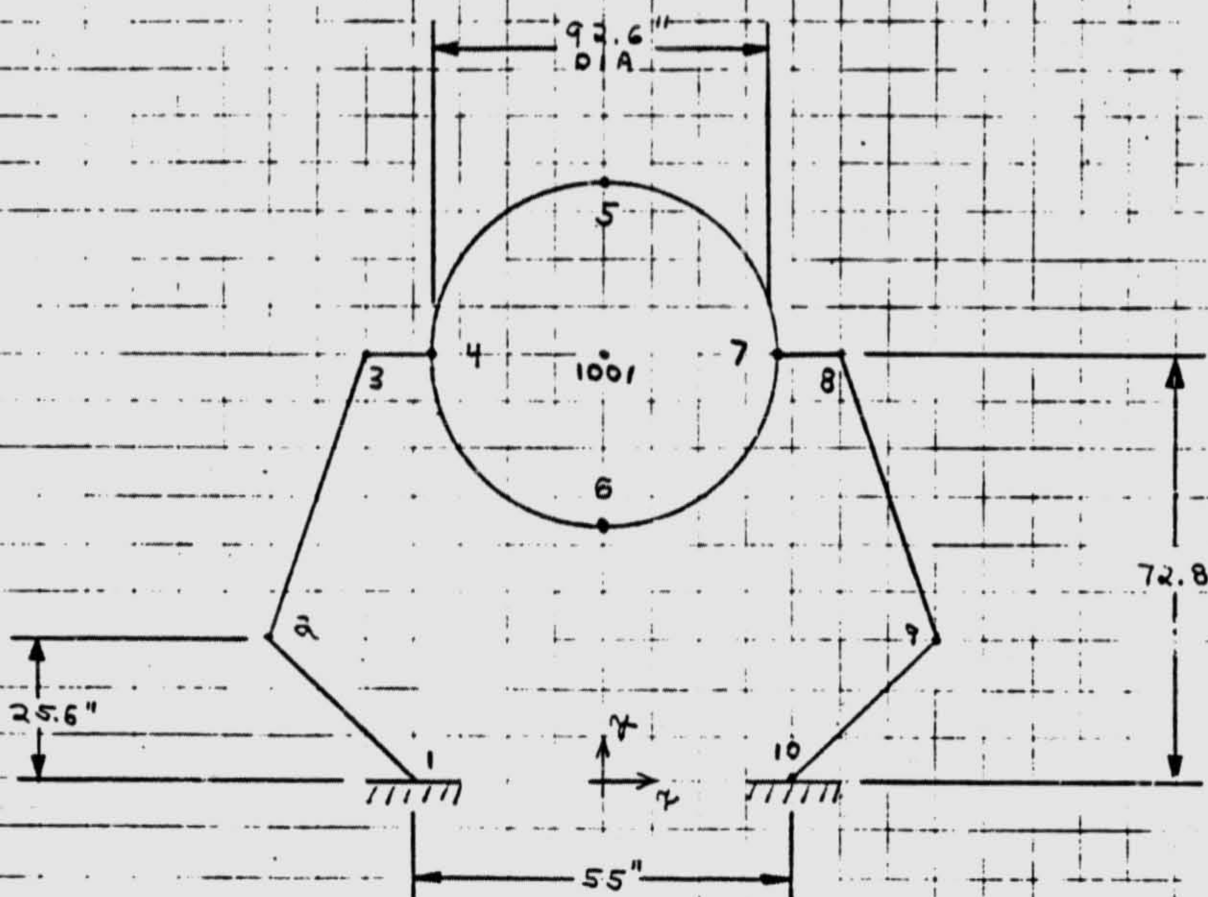
SEE P. 10-14 FOR STRESS CALCULATIONS

TABLE II - ALLOWABLE STRESSES

TYPE	ALUMINUM 7075-T6	STAINLESS STEEL 17-4PH - COND. H900
F_{TU}	76 KSI	190 KSI
F_{TY}	67 KSI	170 KSI
F_{CY}	66 KSI	178 KSI
F_{SU}	45 KSI	123 KSI
E	10×10^6 PSI	29×10^6 KSI
G	3.8×10^6 PSI	11×10^6 KSI
ω	0.101 LB/IN^3	0.282 LB/IN

DISCRIPTION OF FEAST MODEL

GEOMETRY

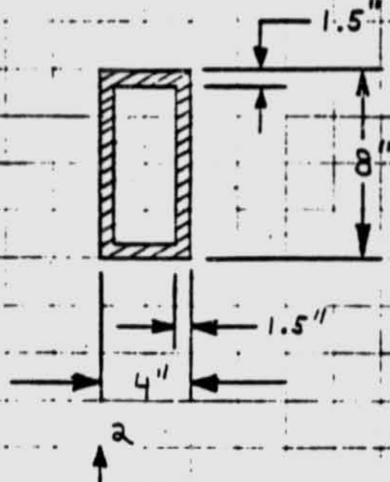


NOTES: 1- NODES 1 & 10 ARE FIXED IN TRANSLATION AND ROTATION
2- NO TORSION TRANSMITED TO ELEMENT 7-8 AT NODE 8
3- ALL LOADS ARE APPLIED IN Z-DIRECTION
4- ELEMENTS 3-4 AND 7-8 ARE STEEL. ALL OTHERS ARE ALUMINUM

SECTION PROPERTIES

RING

ELEMENTS 4-5, 5-7, 7-6, 6-4



$$A = (8 \times 4) - (5 \times 1) = 27.0 \text{ IN}^2$$

$$I_2 = \frac{8 \times 4^3}{12} - \frac{5 \times 1^3}{12} = 42.25 \text{ IN}^4$$

$$I_3 = \frac{4 \times 8^3}{12} - \frac{1 \times 5^3}{12} = 160.25 \text{ IN}^4$$

$$K = \frac{2(1.5)^2(6.5)^2(2.5)^2}{8(1.5) + 4(1.5) - 2(1.5)^2}$$

REF: ROARK 4TH ED P. 196

$$= 88.02 \text{ IN}^4$$

SHAFT

ELEMENTS 3-4, 7-8

$D = 1.76$ IN $\quad R = 0.88$ IN

$$A = \pi (.88)^2 = 2.43 \text{ IN}^2$$

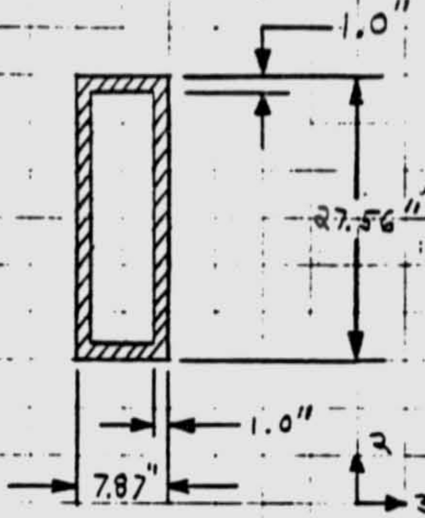
$$I_2 = I_3 = \frac{\pi}{4}(.88)^4 = 0.47 \text{ IN}^4$$

$$K = \frac{\pi}{2}(.88)^4 = 0.94 \text{ IN}^4$$

NOTE: SHAFT IS STEEL; REMAINDER OF STRUCTURE IS ALUMINUM

UPPER YOKE

ELEMENTS 2-3, 8-9



$$A = (27.56 \times 7.87) - (25.56 \times 5.87)$$

$$= 66.86$$

$$I_2 = \frac{27.56 \times 7.87^3}{12} - \frac{25.56 \times 5.87^3}{12}$$

$$= 688.7 \text{ IN}^4$$

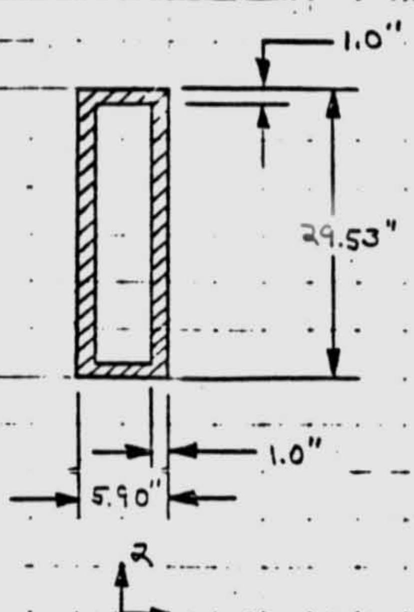
$$I_3 = \frac{7.87 \times 27.56^3}{12} - \frac{5.87 \times 25.56^3}{12}$$

$$= 5560 \text{ IN}^4$$

$$K = \frac{2(1.0)^2(26.56)^2(6.87)^2}{27.56 + 7.87 - 2} = 1991.9 \text{ IN}^4$$

LOWER YOKE

ELEMENTS 1-2, 9-10



$$A = (29.53 \times 5.90) - (27.53 \times 3.9) = 66.86 \text{ IN}^2$$

$$I_2 = \frac{29.53 \times 5.9^3}{12} - \frac{27.53 \times 3.9^3}{12}$$

$$= 369.32 \text{ IN}^4$$

$$I_3 = \frac{5.9 \times 29.53^3}{12} - \frac{3.9 \times 27.53^3}{12}$$

$$= 5879.67 \text{ IN}^4$$

$$K = \frac{2(1.0)^2(28.53)^2(4.9)^2}{29.53 + 5.9 - 2}$$

$$= 1169.2 \text{ IN}^4$$

LOADING

ALL LOADS APPLIED TO FEAST MODEL ARE IN THE Z-DIRECTION AND OF 1g MAGNITUDE.

RING ELEMENTS 4-5, 5-7, 7-6, 6-4

$WT = (27.0)(.101) = 2.7$ LB/IN

APPLIED LOAD = 15.46 LB/IN FROM 4400 LB LOAD

$LOAD = 2.7 + 15.46 = 18.16$ LB/IN

SHAFT ELEMENTS 3-4, 7-8

$LOAD = WT. = (.282)(2.43) = 0.69$ LB/IN ≈ 0 LBS/IN

YOKE ELEMENTS 1-2, 2-3, 8-9, 9-10

$LOAD = WT = (.10)(66.86) = 6.69$ LB/IN.

NATURAL FREQUENCY

THE APROXIMATE NATURAL FREQUENCY OF THIS STRUCTURE CAN BE CALCULATED FROM THE EQUATION FOR A CANTELEVER UNDER END LOAD WHICH IS

$$f_m = \frac{3.13}{\sqrt{\frac{WL^3}{3EI}}} = \frac{3.13}{\sqrt{\delta_{1g}}}$$ REF: ROARK 4TH EDITION P 369

IF WE USE THE AVE OF THE Z-DEFLECTION AT NODE # 5 AND NODE # 6 FOR δ_{1g}

$$\delta_{z_5} = -.1335 \text{ IN} \quad @ \text{ NODE \# 5}$$

$$\delta_{z_6} = -.0769 \text{ IN} \quad @ \text{ NODE \# 6}$$

$$\therefore \delta_{1g} = \frac{.1335 + .0769}{2} = .105 \text{ IN}$$

SINCE THE PRIMARY INERTIA OF THE RING WAS IMPUT INTO THE FEAST PROGRAM IN ERROR, IT IS NECESSARY TO CORRECT THIS FIGURE. THUS, APPLYING A RATIO OF INERTIAS TO THE DEFLECTION BETWEEN NODES # 4 AND # 7 ~

$$\delta_{z_4} = \delta_{z_7} = -0.039 \text{ IN}$$

$$I_{IMPUT} = 106.25 \text{ IN}^4$$

$$I_{TRUE} = 160.25 \text{ IN}^4$$

$$\delta_{1g} = .039 + \frac{106.25}{160.25}(.105 - .039) = 0.083 \text{ IN}$$

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CALCULATING THE NATURAL FREQUENCY ~

$$f_m = \frac{3.13}{\sqrt{\delta_{1g}}} = \frac{3.13}{\sqrt{.083}}$$

$$\underline{\underline{f_m = 10.9 \text{ HERTZ}}}$$

THIS IS ACCEPTABLE SINCE IT IS GREATER THAN THE REQUIRED NATURAL FREQUENCY OF 5.0 HERTZ.

IT SHOULD BE NOTED THAT ALTHOUGH THE INCORRECT RING INERTIA HAS A SIGNIFICANT EFFECT ON THE NATURAL FREQUENCY, IT WILL NOT HAVE A SIGNIFICANT EFFECT ON THE MEMBER INTERNAL LOADS DUE TO THE NATURE OF THE STRUCTURE.

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STRESS ANALYSIS

RING ELEMENTS 4-5, 5-7, 7-6, 6-4

ELEM	END	F1	F2	F3	M4	M5	M6
4-5	4	0	-1339	0	-525	0	39,100
	5	0	-10	0	307	0	23,220
5-7	5	0	10	0	-307	0	-23,220
	7	0	-1331	0	-510	0	-39,100
7-6	7	0	-1339	0	-525	0	39,100
	6	0	-10	0	307	0	23,220
6-4	6	0	10	0	-307	0	-23,220
	4	0	-1331	0	-510	0	-39,100

CHECKING BENDING AT EITHER NODE #4 OR NODE #7

$$M_{ULT} = 9 \times 39{,}100 = 351{,}900 \text{ IN-LBS}$$

$$f_B = \frac{(351{,}900)(4)}{160.25} = 8784 \text{ PSI}$$

$$F_{TU} = 76{,}000 \text{ PSI} \quad \text{FOR } 7075\text{-T6}$$

$$MS = \frac{76{,}000}{8{,}784} - 1 = \underline{\underline{+\ 7.65}}$$

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CHECKING COMBINED SHEAR AND TORSION AT ELEMENT #7-6, NODE #7 END OR ELEMENT #4-5, NODE #4 END

$$T_{ULT} = 9 \times 525 = 4,725 \text{ IN-LBS}$$

$$V_{ULT} = 9 \times 1339 = 12,051 \text{ LBS}$$

$$f_{ST} = \frac{4725}{2 \times 1.5 \times (8-1.5)(4-1.5)} = 97 \text{ PSI}$$

$$f_S = \frac{12,051}{27.0} = 446 \text{ PSI}$$

$$f_{S_{TOT}} = 97 + 446 = 543 \text{ PSI}$$

$$F_{SU} = 45,000 \text{ PSI} \quad \text{FOR } 7075\text{-T6}$$

$$MS = \frac{45,000}{543} - 1 = \underline{\underline{\text{AMPLE}}}$$

∴ RING IS ACCEPTABLE

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SHAFT ELEMENTS 3-4, 7-8

ELEM	END	F1	F2	F3	M4	M5	M6
3-4	3	0	-2670	0	0	0	9378
	4	0	2670	0	0	0	1035
7-8	7	0	2670	0	0	0	-1035
	8	0	-2670	0	0	0	-9378

CHECKING BENDING AT EITHER NODE # 3 OR NODE # 8

$M_{ULT} = 9 \times 9378 = 84,402$ IN-LBS

$$f_B = \frac{(84,402) \times (.88)}{0.47} = 158,029 \text{ PSI}$$

$F_{TU} = 190$ FOR 17-4 PH STEEL H900 CONDITION

$$MS = \frac{190}{158} - 1 = \underline{\underline{+.20}}$$

SHEAR STRESS WILL NOT BE CRITICAL

∴ SHAFT IS ACCEPTABLE

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UPPER YOKE ELEMENTS 2-3, 8-9

ELEM	END	F1	F2	F3	M4	M5	M6
2-3	2	0	-2986	0	9378	0	133500
	3	0	2670	0	-9378	0	0
8-9	8	0	2670	0	-9378	0	0
	9	0	-2986	0	9378	0	-133,500

CHECKING BENDING AT EITHER NODE # 2 OR NODE # 9 ~

$$M_{ULT} = 9 \times 133{,}500 = 1{,}201{,}500 \text{ IN-LBS}$$

$$f_B = \frac{(1{,}205{,}500) \times \left(\frac{27.56}{2}\right)}{5560} = -2978 \text{ PSI}$$

$F_{TU} = 76{,}000$ PSI FOR 7075-T6

$$MS = \frac{76{,}000}{2{,}978} - 1 = \underline{\underline{\text{AMPLE}}}$$

CHECKING COMBINED SHEAR AND TORSION AT EITHER NODE # 2 OR NODE # 9 ~

$T_{ULT} = 9 \times 9378 = 84{,}402$ ''# $\quad V_{ULT} = 9 \times 2986 = 26{,}874$ #

$$f_S = \frac{84{,}402}{2 \times 1 \times (27.56 - 1)(7.87 - 1)} + \frac{26{,}874}{66.86}$$

$$= 231 + 402$$

$$= 633 \text{ PSI}$$

$F_{SU} = 45{,}000$ PSI FOR 7075-T6

$$MS = \frac{45{,}000}{633} - 1 = \underline{\underline{\text{AMPLE}}}$$

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LOWER YOKE   ELEMENTS 1-2, 9-10

ELEM	END	F1	F2	F3	M4	M5	M6
1-2	1	0	-3219	0	97,560	0	199,800
	2	0	2986	0	-97,560	0	-96,530
9-10	9	0	2986	0	-97,560	0	91,580
	10	0	-3219	0	97,560	0	-199,800

CHECKING BENDING AT EITHER NODE # 1 OR NODE # 10 ~

$M_{ULT} = 9 \times 199{,}800 = 1{,}798{,}200$ IN-LBS

$$f_B = \frac{(1{,}798{,}200) \times \left(\frac{29.53}{2}\right)}{5879.67} = 4515 \text{ PSI}$$

$F_{TU} = 76{,}000$ PSI   FOR 7075-T6

$$MS = \frac{76{,}000}{4515} - 1 = \underline{\underline{\text{AMPLE}}}$$

CHECKING COMBINED SHEAR AND TORSION AT EITHER NODE # 1 OR NODE # 10 ~

$T_{ULT} = 9 \times 97{,}560 = 878{,}040$ "# $V_{ULT} = 9 \times 3219 = 28{,}971$ #

$$f_S = \frac{878{,}040}{2 \times 1 \times (29.53-1)(5.9-1)} + \frac{28{,}971}{66.86}$$

$$= 3141 + 433 = 3574 \text{ PSI}$$

$F_{SU} = 45{,}000$ FOR 7075-T6

$$MS = \frac{45{,}000}{3574} - 1 = \underline{\underline{\text{AMPLE}}}$$

∴ YOKE IS ACCEPTABLE

FORM 1-8136 B (9-58)

BY C. FAUST CK. DATE 12/8/78 REV.	GENERAL ELECTRIC ORIGINAL PAGE IS OF POOR QUALITY	PAGE MODEL FIM REPORT

INTERFACE BOLT LOADS (Pallet Hardpoints)

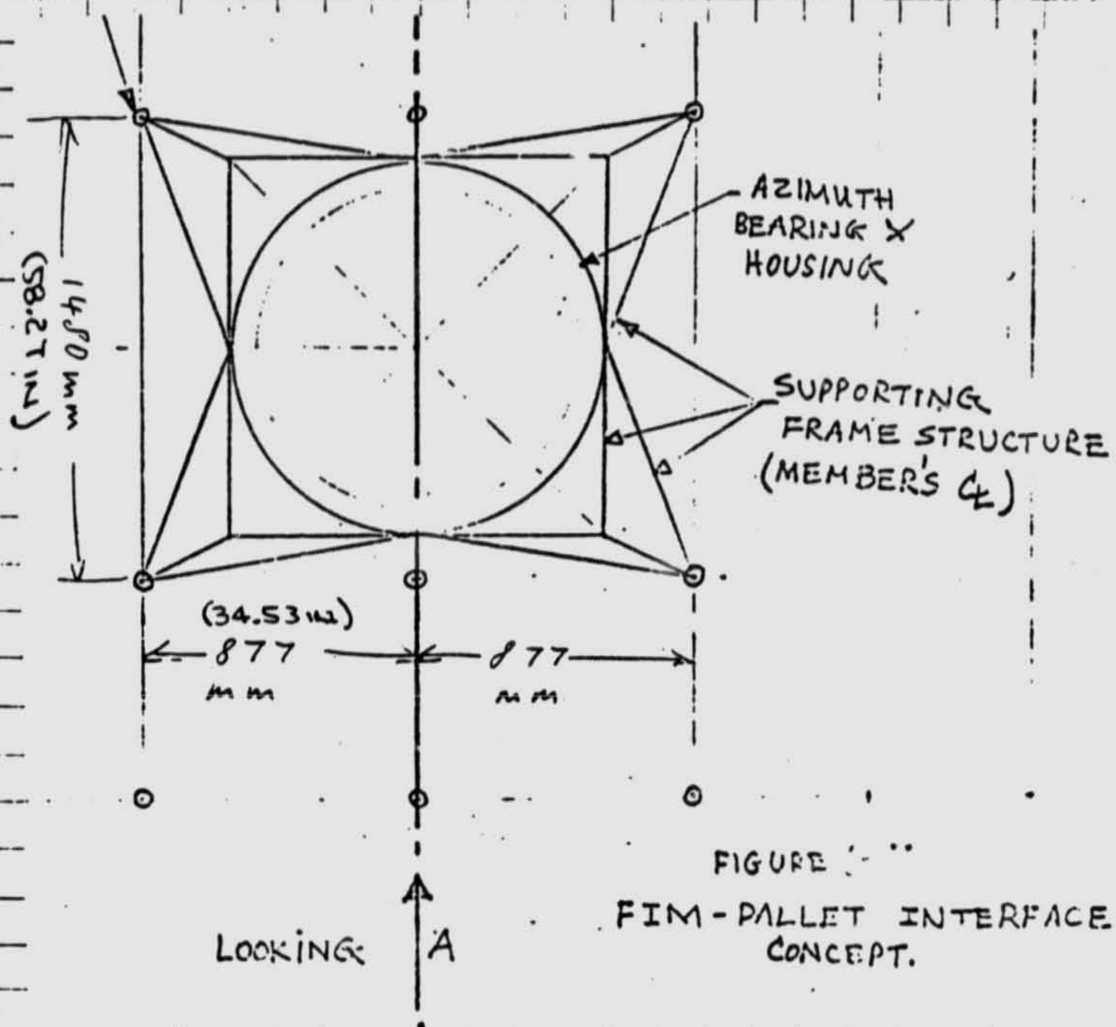

FIGURE :

FIM-PALLET INTERFACE CONCEPT.

FOR LOADING IN THE - DIRECTION THE FIM/PALLET INTERFACE BOLTS WILL REACT THE OVERTURNING MOMENT IN TENSION. THE MAGNITUDE BEING:

$$P = \frac{1}{3}\,\frac{4400^{\#}(9g\text{'s})(72.8'')}{58.27} = \underline{16492.}\ \text{LBS (ULT)}$$

$$13199.\ \text{LBS (YIELD)}$$

FORM 1-8138 E (9-58)

BY C. FAUST	GENERAL ELECTRIC	PAGE
CK.		MODEL FJM
DATE 12/8/78 REV.		REPORT

THE SHEAR BEING REACTING BY ALL 6 BOLTS

$$\therefore \quad V = \frac{1}{6}(4400)(9) = \underline{6600.} \text{ LBS (ULT)}$$
$$5280. \text{ LBS (YIELD)}$$

Note: Ultimate hardpoint/pallet load capabilities for the hardpoints used are given in SVAH, Appendix B, Page B.4.1-23 (CDR Issue). For a discussion of actual hardpoint capabilities refer back to Section 5.2

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FORM 1-8133 B (9-58)

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APPENDIX A ~ FEAST INPUT

```
F 1     F I M  MODEL    10 LOAD    PASS 3
0 0 0 0 0
11 10 0 2 1 -1
1 -27.5 0 0
2 -51.2 25.6 0
3 -51.2 72.8 0
4 -47.3 72.8 0
5 0 119.1 0
6 0 26.5 0
7 47.3 72.8 0
8 51.2 72.8 0
9 51.2 25.6 0
10 27.5 0 0
1001 0 72.8 0
1 2 1001 66.86 1E7 3.8E6 1169.2 2 2 369.32 5879.67 0 0 0 0 6.69 0 0 0 0
0
2 3 1001 66.86 1E7 3.8E6 1991.9 2 2 688.7 5560.0 0 0 0 0 6.69 0 0 0 0
0
3 4 1 2.4 29E6 11E6 .94 2 2 .471 .471 0 0 0 0 0 0 0 0 0
0
4 5 -1001 27 1E7 3.8E6 88.02 1.11 1.11 42.25 106.25 0 0 0 0 18.16 0 0 0 0
0
5 7 -1001 27 1E7 3.8E6 88.02 1.11 1.11 42.25 106.25 0 0 0 0 18.16 0 0 0 0
0
7 6 -1001 27 1E7 3.8E6 88.02 1.11 1.11 42.25 106.25 0 0 0 0 18.16 0 0 0 0
0
6 4 -1001 27 1E7 3.8E6 88.02 1.11 1.11 42.25 106.25 0 0 0 0 18.16 0 0 0 0
0
7 8 10 2.4 29E6 11E6 .94 2 2 .471 .471 0 0 0 0 0 0 0 0 0
-1
0 0 0 0 0 0 0 0 0 1 0 0
9 9 1001 66.86 1E7 3.8E6 1991.9 2 2 663.7 5560.0 0 0 0 0 6.69 0 0 0 0
0
9 10 1001 66.86 1E7 3.8E6 1169.2 2 2 369.32 5879.67 0 0 0 0 6.69 0 0 0 0
0
1 1 0 1 0 1 0 1 0 1 0 1 0
10 1 0 1 0 1 0 1 0 1 0 1 0
```

RESULTS FOR LOAD CASE 1

DEFLECTIONS

JOINT NO	X (IN)	Y (IN)	Z (IN)	OX (RAD)	OY (RAD)	OZ (RAD)
1	0.	0.	0.	0.	0.	0.
2	1.982E-14	1.823E-14	-2.540E-03	-5.835E-04	5.037E-04	-1.406E-15
3	9.930E-14	1.824E-14	-3.289E-0[illegible]	-6.391E-04	5.622E-04	-1.786E-15
4	9.944E-14	1.354E-14	-3.916E-02	-6.391E-04	1.753E-03	-6.716E-16
5	1.713E-13	-4.904E-14	-1.335E-01	-2.109E-03	-1.264E-05	-2.958E-16
6	1.202E-13	3.464E-14	-7.690E-02	8.872E-04	1.264E-05	-2.994E-16
7	1.921E-13	-2.705E-14	-3.916E-02	-5.826E-04	-1.753E-03	-7.633E-16
8	1.918E-13	-3.518E-14	-3.289E-02	-6.391E-04	-5.622E-04	-3.447E-15
9	3.823E-14	-3.517E-14	-2.540E-03	-5.835E-04	-5.037E-04	-2.716E-15
10	0.	0.	0.	0.	0.	0.

MEMBER INTERNAL LOADS

	FX F1 (LB)	FY F2 (LB)	FZ F3 (LB)	MX M4 (IN-LB)	MY M5 (IN-LB)	MZ M6 (IN-LB)
MEMBER 1 I= 1 J= 2						
I=	-2.524E-09	-1.350E-10	3.219E 03	2.129E 05	6.415E 04	1.828E-07
J=	2.524E-09	1.350E-10	-2.986E 03	-1.335E 05	9.378E 03	-1.150E-07
I=	-1.616E-09	-3.219E 03	-1.944E-09	9.756E 04	-1.828E-07	1.998E 05
J=	1.616E-09	2.986E 03	1.944E-09	-9.756E 04	1.150E-07	-9.158E 04
MEMBER 2 I= 2 J= 3						
I=	-2.524E-09	-1.350E-10	2.986E 03	1.335E 05	-9.378E 03	1.150E-07
J=	2.524E-09	1.350E-10	-2.670E 03	4.114E-03	9.378E 03	4.166E-09
I=	1.350E-10	-2.986E 03	-2.524E-09	9.378E 03	-1.150E-07	1.335E 05
J=	-1.350E-10	2.670E 03	2.524E-09	-9.378E 03	-4.166E-09	4.114E-03
MEMBER 3 I= 3 J= 4						
I=	-2.524E-09	-1.350E-10	2.670E 03	1.521E-03	-9.378E 03	-4.166E-09
J=	2.524E-09	1.350E-10	-2.670E 03	-1.521E-03	-1.035E 03	3.640E-09
I=	2.524E-09	-2.670E 03	1.350E-10	-1.521E-03	4.166E-09	9.378E 03
J=	-2.524E-09	2.670E 03	-1.350E-10	1.521E-03	-3.640E-09	1.035E 03
MEMBER 4 I= 4 J= 5						
I=	-1.300E-09	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

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J=	1.299E-09	6.756E-11	1.042E 01	-3.070E 02	-2.322E 04	2.114E-08
I=	6.756E-11	-1.339E 03	-1.299E-09	-5.249E 02	-3.583E-08	3.910E 04
J=	-1.299E-09	-1.042E 01	-6.756E-11	3.070E 02	-2.114E-08	2.322E 04
MEMBER 5 I= 5 J= 7						
I=	-1.299E-09	-6.756E-11	-1.042E 01	3.070E 02	2.322E 04	-2.113E-08
J=	1.299E-09	6.756E-11	1.331E 03	3.910E 04	-5.098E 02	-4.222E-08
I=	1.299E-09	1.042E 01	6.756E-11	-3.070E 02	2.113E-08	-2.322E 04
J=	6.756E-11	-1.331E 03	-1.299E-09	-5.098E 02	4.222E-08	-3.910E 04
MEMBER 6 I= 7 J= 6						
I=	1.225E-09	6.748E-11	1.339E 03	-3.910E 04	-5.249E 02	3.308E-08
J=	-1.225E-09	-6.748E-11	1.042E 01	3.070E 02	2.322E 04	2.043E-08
I=	6.748E-11	-1.339E 03	-1.225E-09	-5.249E 02	-3.308E-08	3.910E 04
J=	-1.225E-09	-1.042E 01	-6.748E-11	3.070E 02	-2.043E-08	2.322E 04
MEMBER 7 I= 6 J= 4						
I=	1.225E-09	6.748E-11	-1.042E 01	-3.070E 02	-2.322E 04	-2.043E-08
J=	-1.225E-09	-6.748E-11	1.331E 03	-3.910E 04	5.098E 02	-3.947E-08
I=	1.225E-09	1.042E 01	6.748E-11	-3.070E 02	2.043E-08	-2.322E 04
J=	6.748E-11	-1.331E 03	-1.225E-09	-5.098E 02	3.947E-08	-3.910E 04
MEMBER 8 I= 7 J= 8						
I=	-2.524E-09	-1.350E-10	-2.670E 03	0.	1.035E 03	9.135E-09
J=	-4.927E-09	1.350E-10	2.670E 03	0.	9.378E 03	-9.661E-09
I=	2.524E-09	2.670E 03	1.350E-10	0.	-9.135E-09	-1.035E 03
J=	4.927E-09	-2.670E 03	-1.350E-10	0.	9.661E-09	-9.378E 03
MEMBER 9 I= 8 J= 9						
I=	4.927E-09	-1.350E-10	-2.670E 03	-4.880E-02	-9.378E 03	9.661E-09
J=	-4.927E-09	1.350E-10	2.986E 03	1.335E 05	9.378E 03	2.229E-07
I=	-1.350E-10	2.670E 03	-4.927E-09	-9.378E 03	-9.661E-09	4.880E-02
J=	1.350E-10	-2.986E 03	4.927E-09	9.378E 03	-2.229E-07	-1.335E 05
MEMBER 10 I= 9 J= 10						
I=	4.927E-09	-1.350E-10	-2.986E 03	-1.335E 05	-9.378E 03	-2.229E-07
J=	-4.927E-09	1.350E-10	3.219E 03	2.129E 05	-6.415E 04	3.522E-07
I=	3.248E-09	2.986E 03	-3.707E-09	-9.756E 04	2.229E-07	9.158E 04
J=	-3.248E-09	-3.219E 03	3.707E-09	9.756E 04	-3.522E-07	-1.998E 05

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SUMMATION OF FORCES

JOINT NO	FX (LB)	FY (LB)	FZ (LB)	MX (IN-LB)	MY (IN-LB)	MZ (IN-LB)
1	-2.524E-09	-1.350E-10	3.219E 03	2.129E 05	6.415E 04	1.828E-07

2	1.721E-15	-4.335E-15	2.747E-04	2.143E-02	1.782E-02	1.776E-15
3	1.060E-14	4.493E-16	-6.104E-05	5.635E-03	6.104E-04	5.052E-15
4	1.274E-14	1.466E-16	5.798E-04	0.	3.128E-04	7.105E-15
5	-3.331E-16	7.286E-16	-3.862E-05	-1.026E-03	-7.324E-04	1.865E-14
6	-2.498E-16	1.717E-16	-5.281E-05	1.919E-03	-4.883E-04	1.332E-15
7	-4.829E-15	6.592E-17	1.221E-04	-4.883E-04	-2.869E-03	7.772E-15
8	5.174E-14	-3.031E-15	-2.136E-04	-4.880E-02	-4.883E-04	1.033E-14
9	4.996E-16	1.146E-14	-1.312E-03	-9.766E-03	-3.784E-03	3.020E-14
10	-4.927E-09	1.350E-10	3.219E 03	2.129E 05	-6.415E 04	3.522E-07

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APPENDIX B

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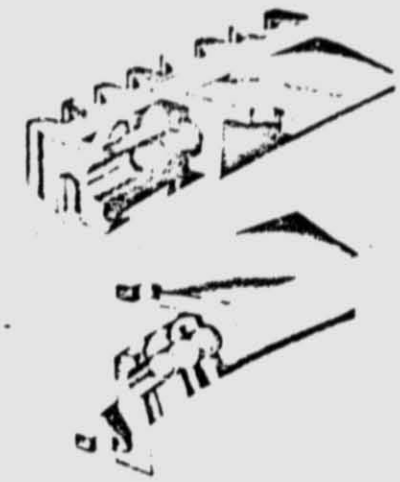

Series 2000

Series 1000

The Econo-Trak model-numbering system

L = Series 2000
M = Series 1000
First number = Approximate diameter of ball, in eighths of an inch
Second number = Approximate diameter of raceway, in inches
N = Internal gear
E = External gear
P = Gearless
Z = Mounting holes included

Examples:

Model L9-38E1Z indicates a Series 2000 bearing with approximately 1 1/8" balls, approximately 38" raceway, external gear, mounting holes.

Model M5-22P1 indicates a Series 1000 bearing with approximately 5/8" balls and approximately 22" raceway. All Series 1000 bearings are designated P1, for they are available only gearless. Mounting holes (Z) are an optional extra. See page 21.

For recommendations by a Rotek applications specialist on the selection and use of big bearings, you are invited to call our headquarters (216-296-9951) and ask for Extension 223.

ROTEK

Rotek Incorporated
220 West Main Street
Ravenna, Ohio 44266

. . . the Big Bearing people

Load calculation for extraordinary applications

For applications of combined loading which do not meet the limitations of the simplified selection charts on pages 13 and 15, or for any application as desired by the designer, Econo-Trak bearings may be selected according to the following formula. Note that all loadings are converted to equivalent thrust loading by means of simple coefficients. It is necessary to assume a race diameter (D) in order to complete the calculation. Results must be checked by recalculating with the actual race diameter of the selected bearing. Bearing race diameters and thrust capacity values are listed in the rating tables on pages 12-13.

Equivalent thrust load (A_{eq}) =

Actual thrust load (A)

$$+ \frac{4.37 \times \text{moment load (M)}}{\text{Raceway dia. (D) (in feet)}}$$

$+ 3.44 \times$ radial load (R)

Summarizing:

$$A_{eq} = A + \frac{4.37\ M}{D} + 3.44\ R$$

(Note that moment load is the algebraic sum of Ae + Re')

The equivalent thrust load, as produced by this calculation, is defined as the thrust load which, if applied, would cause the same loading of the most heavily loaded balls as that which occurs under the actual conditions of loading. After determining the equivalent thrust load, the bearing may be selected directly from the thrust capacity column on page 12. If bearing rotation is less than one hour per day and peripheral speeds of less than 100 ft per minute, bearing may be selected according to the brown capacity figures. If higher speed or more frequent or continuous rotation is required, dynamic capacity will be the determining consideration. Please see discussion of dynamic capacity and dynamic capacity calculations on pages 9 to 11 before performing dynamic capacity calculations. Note Example 4 on page 17.

Frictional torque

Frictional torque, or the force required to rotate a fully loaded bearing, can be considerable, and must be taken into account when designing a power supply and gear train.

An approximate calculation of frictional torque can be made using the formula shown below. The formula is based on classic bearing theory, not on actual test experience, which varies widely with the application.

It should be noted that torque values obtained from this equation are very rough approximations, and that actual torque, even among supposedly identical bearings under identical service conditions, may vary by as much as five times. In addition, rigidity and accuracy of the mounting structure greatly affect the frictional torque. Therefore, it is recommended that a service factor of at least 5 be employed in determining required drive torque so that sufficient power will be available to rotate a fully loaded bearing under adverse conditions.

$$T = .003\ (4.4M + AD + 2.2RD)$$

when

T = torque (lb-ft)
M = moment load (ft-lb)
A = thrust load (lb)
R = radial load (lb)
D = raceway diameter (ft)

Gear torque capacity

Tangential tooth-load capacities listed on page 19 are based on the Lewis equation at 18,500psi root stress. For low-speed intermittent rotation, calculate gear torque capacity by multiplying maximum tangential tooth load by pitch radius (half of dimension PD). For continuous rotation, check dynamic capacity of gears by using AGMA formulas. Make certain in all cases that drive pinion has adequate tooth capacity. If pinion has twelve or more teeth and minimum hardness of 250 BHN, pinion tooth strength will equal or exceed that of bearing gear.

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Dynamic capacity calculations

The dynamic capacities listed in the rating charts provide theoretical B_{10} life of one million revolutions, which equals theoretical median life (B_{50}) of five million revolutions. Bearing life is inversely proportional to the cube of the applied load. In order to select bearings for life expectancies of other than one million revolutions, the applied load is adjusted as follows:

$$\text{Load adjustment factor, f} = \sqrt[3]{\frac{\text{Desired life}}{\text{1,000,000 revolutions}}}$$

Table D1 lists the f (cube root) values for a variety of desired life conditions. Note that life calculations for fewer than 600,000 revolutions are not considered reliable. For all applications requiring life of less than 600,000 revolutions, the bearing should be selected on the basis of 600,000 revolutions. *Do not exceed the intermittent capacity rating.* Table D2 and D3 list the number of hours of operation which may be expected at various operating speeds, and also the number of hours per year under various operating conditions, assuming that the bearing is continuously rotating during the operation of the equipment.

Table D1

Desired life (B_{10}) in revolutions	Load-adjustment factor (f)
600,000	0.843†
700,000	0.888
800,000	0.928
900,000	0.965
1,000,000	1.00
1,250,000	1.08
1,500,000	1.15
2,000,000	1.26
3,000,000	1.44
4,000,000	1.59*
5,000,000	1.71*
7,500,000	1.96*
10,000,000	2.15*
15,000,000	2.46*
20,000,000	2.71*
25,000,000	2.92*
30,000,000	3.11*
50,000,000	3.68*

*Life factors are based upon theoretical fatigue life of raceways and balls. Ball separators (spacers) are subject to frictional wear and may require periodic replacement in long-life applications. Accuracy of mounting structure, speed, and quality of lubrication will influence actual spacer life.

†Minimum value for reliable life calculation.

Table D2

Theoretical bearing life at various operating speeds based on B_{10} life of 1,000,000 revolutions

Speed (RPM)	Hours
1	16,700
2	8,350
5	3,340
10	1,670
20	835
50	334
100	167

Table D3

Number of hours resulting from various service conditions (continuous rotation during working hours)

Single-shift Operation	40 hr/wk x 50 wk/yr. = 2000 hr/yr.
Three-shift Industrial Operation	120 hr/wk x 50 wk/yr. = 6000 hr/yr.
Continuous 24-hour Operation	168 hr/wk x 52 wk/yr. = 8750 hr/yr.

Life calculation example 1

Speed = 10 RPM = 600 revolutions per hour.
Usage = 24 hours per day continuous (8750 hr/yr.).
Desired B_{10} life = 5 years.
Number of bearing revolutions during desired life: 600 revolutions/hr x 8750 hr/yr. x 5 yrs. = 26,200,000 revolutions.

$$\text{Load adjustment factor} = f = \sqrt[3]{\frac{\text{Desired life}}{\text{1,000,000 revolutions}}} = \sqrt[3]{\frac{26{,}200{,}000}{1{,}000{,}000}} = \sqrt[3]{26.2} = 2.97$$

(This value could have been estimated with sufficient accuracy from Table D1.)

Thus, before selecting a bearing from dynamic capacity ratings, multiply actual load by 2.97.

Life calculation example 2

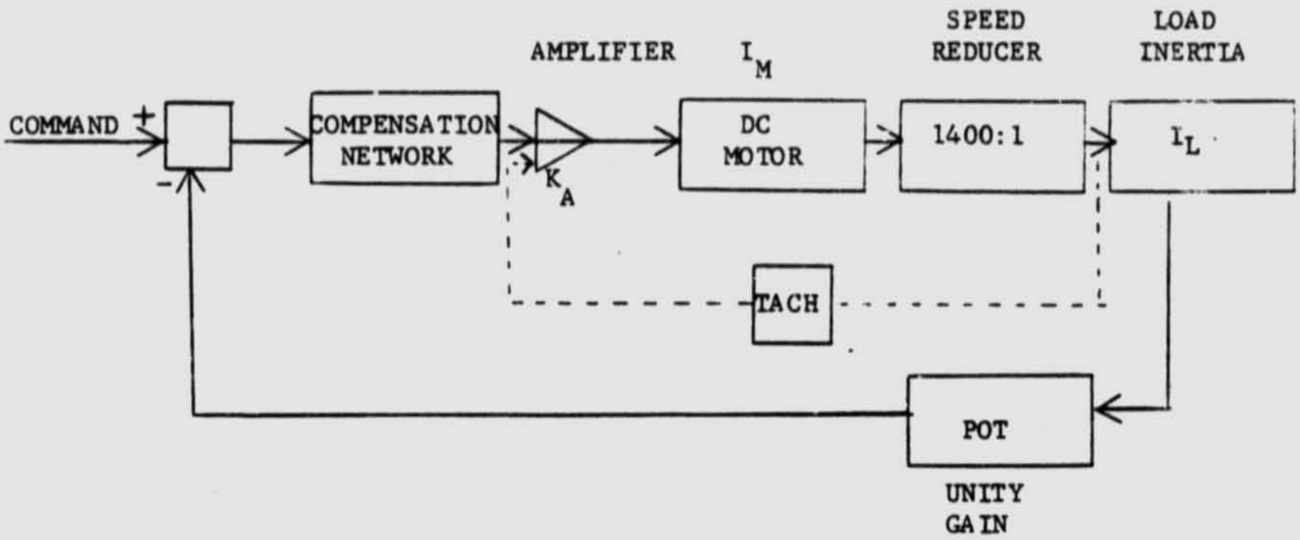
Speed = 2 RPM = 120 revolutions per hour.
Usage = 1 hr per day, 5 days per wk = 250 hr/yr.
Desired B_{10} life = 5 years.
Number of bearing revolutions during desired life: 120 revolutions/hr x 250 hr/yr. x 5 years = 150,000 revolutions.

Note that 150,000 revolutions is below the 600,000 minimum for accurate life calculations. Load adjustment factor should be based on 600,000 revolutions which, referring to Table D1, is .843. Actual bearing loads may be multiplied by .843 before selecting bearing from dynamic capacity rating.

APPENDIX C

FLEXIBLE INSTRUMENT MOUNT SERVO SYSTEM

FLEXIBLE INSTRUMENT MOUNT SERVO SYSTEM



PROBLEM: To specify control system needed to establish and maintain FIM orientation in 2 axes relative tothe orbiter within a 1° accuracy when the orbiter is in circular orbit.

MOTOR, LOAD INERTIA, GEAR REDUCER

Inertia (as seem at load)

I_{load} (Azimuth) = 1,000 slug ft^2

I_{load} (Elevation) = 850 slug ft^2

Includes load, gimbals and structures

$I_{motor} = 1.8 \times 10^{-4}$ lb.ft $sec^2 = 1.8 \times 10^{-4}$ slug ft^2

NOTE: Because this problem is, to some extent, dominated by non-viscous (i.e., constant) friction, it cannot be analyzed by standard linear techniques. Hence, a brief investigation can only provide a very preliminary concept of a system design.

$$I_{T(\Delta Z)} = 1{,}000 + 1.8 \times 10^{-4} \times (1400)^2 = 1{,}353 \text{ slug ft.}^2$$
$$I_{T(El)} = 850 + 1.8 \times 10^{-4} \times (1400)^2 = 1{,}203 \text{ slug ft.}^2$$

Inertia load as seen at load

Friction (Az) = 30 ft. lbs.

Friction (El) = 1.6 ft. lbs. (as seen at load)

Frictions at 1g
Estimated to be reduced by factor of 50 at 0g

RCS MAX ANGULAR ACCEL. LEVELS: (FROM PAYLOAD ACCOMMODATION HANDBOOK):

	$\pm\ddot{\phi}$ deg/sec^2	$+\ddot{\theta}$	$-\ddot{\theta}$	$\pm\ddot{\psi}$
Primary Thrusters →	1.2	1.4	1.5	0.8
Secondary Thrusters →	0.04	0.03	0.02	0.02

<u>ASSUME</u>: Control need only be concerned with operation when <u>secondary</u> thrusters are operational (and not primary thrusters) also assume worst case accel. of 0.04 $^\circ$/sec^2.

Torque needed to hold gimbal during thrusting:

$$T = I\alpha$$

<u>Azimuth</u>:

$$T = 1{,}353 \text{ slug ft.}^2 \times 0.04\ ^\circ/\text{sec}^2 \times \frac{\pi}{180} = 0.94 \text{ ft. lbs.}$$

<u>NOTE</u>: This compares with (non-viscous) friction torque of <u>30 ft. lbs. at 1g conditions</u>.

<u>Elevation</u>:

$$T = 1{,}203 \text{ slug ft.}^2 \times 0.04\ ^\circ/\text{sec}^2 \times \frac{\pi}{180} = 0.84 \text{ ft. lbs.}$$

<u>NOTE</u>: This compares with (non-viscous) friction torque of 1.6 ft. lbs. <u>at 1g conditions</u>.

<u>Conclusion</u>: for operation at 1g:

If gimbal axes are principle inertial axes then friction torque alone is sufficient to hold gimbal. (More than enough in elevation by factor of ~30 in azimuth and

factor of ~ 2 in elevation).

Conclusion: For steady state operation at 1g servo control can be turned off.

NOTE: Above conclusion is true for the assumed friction levels. If friction levels are less than these levels it may be desireable to either employ a locking mechanism or to continue to energize the servo drive. Alternatively if the orbiter has limit cycle amplitudes less than ($\pm$?) 1° accuracy requirements then it may be acceptable to allow this angular motion to occur at gimbals.

The above conclusion is significant only if there is a plan for simulating on the ground the limit cycle attitude oscillation of the orbiter. Since bearing friction at 1g is so much greater than at 0g such a simulation may not be of interest.

OPERATION AT 0g

Friction Torques at 0g are estimated to be 1/50 of those at 1g or

Azimuth: 0.6 ft. lbs.

Elevation: 0.03 ft. lbs.

NOTE: Hence friction will supply 2/3 torque needed in azimuth but only a negligible of that needed in elevation.

Therefore, servos must be energized in space, or alternatively, a locking mechanism will be needed. This is true, however, only when the amplitude of attitude limit cycle of the orbiter exceeds 1° orientation accuracy requirement.

If it is necessary to use servos in holding function then the required average angular torque is:

$I \ddot{\alpha} R$. where R is ACS Duty Cycle Ratio

	R=10%	R=2%	R=1%
Azimuth:	0.09 ft. lbs.	0.018 ft. lbs.	0.009 ft. lbs.
Elevation:	0.08 ft. lbs.	0.016 ft. lbs.	0.008 ft. lbs.

Assumes Worst Case $\ddot{\mathcal{L}}$ = 0.04 ^{o}/sec^{2}

ACS duty cycles for the orbiter are not provided (TBD) in the Payload Accommodation Handbook but can be expected to be less than 1%. Conservatively we will assume an average torque requirement of 0.01 ft. lbs. This is 1/52 of the rated stall torque of the T 2967 motor. Power required at this stall torque is 67.5 watts. Therefore, to hold gimbals with the Torque motor (based on all above assumption) is:

$$\frac{67.5}{52}\text{ watts} = \underline{1.3\text{ watts}}$$

Because the finite azimuth bearing friction does exist however, this average power can be reduced. The estimated holding power then becomes:

<u>Azimuth</u>	<u>Elevation</u>
0.43 watts	1.3 watts

D.C. MOTOR RESPONSE

Typical open loop linear response of the DC Motors plus inertial and friction load is given* by:

$$G(s) = \frac{K}{S(\tau S+1)} \quad \text{i} \tau \approx \frac{I}{F}$$

where I is equivalent moment of inertia seen at load and F is equivalent coefficient of (viscous) friction seen at load; and where effect of back emf on time constant is neglected. For the present ease, however, the viscous friction coefficient is not given and is indicated as being dominated by constant (non-viscous) friction.

*e.g., "Basic Automatic Control Theory" Murphy, Van Nostrand, NY, pg. 58.

Nevertheless physical considerations suggest that there will be some time constant associated with viscous friction or other energy disapative function. Typically, such a time constant can be expected to be smaller than 0.1 sec and greater than 0.01 sec. We will make an assumption of a 0.05 sec time constant.

The forward loop gain will relate some output torque to some input angular error. Thus

$$K_F = \frac{T}{E_0}$$

where K_F is forward loop gain. T is torque measured at the load and E_0 is the angular position error also measured at the load. For error budget purposes we will assume that steady state angular error should not exceed 0.5° for both azimuth and elevation when operating at 0_g.

Based on previously discussed friction torques this leads to a minimum gain K_F as follows:

$$\text{Azimuth:} \quad K_F = \frac{0.6}{0.5^{\circ}} = 1.2 \text{ ft. lbs./degree}$$

$$\text{Elevation:} \quad K_F = \frac{0.032}{0.5} = 0.064 \text{ ft. lbs./degree}$$

Furthermore it is assumed that this response is needed at some frequency which is high enough to compensate for the limit cycle attitude acceleration disturbances.

A preliminary judgement is made that this gain should exist at a frequency of 10 radians/sec. This should be ample to compensate sufficiently rapidly for the vehicle attitude impulse disturbances.

The response of the azimuth and elevation is shown in the attached attenuation plot. Based on an assumption of an 0.05 sec motor/load time constant as discussed above it is seen that the predicted performance for the relatively low gain systems should be stable without addition of further stabilization loops.

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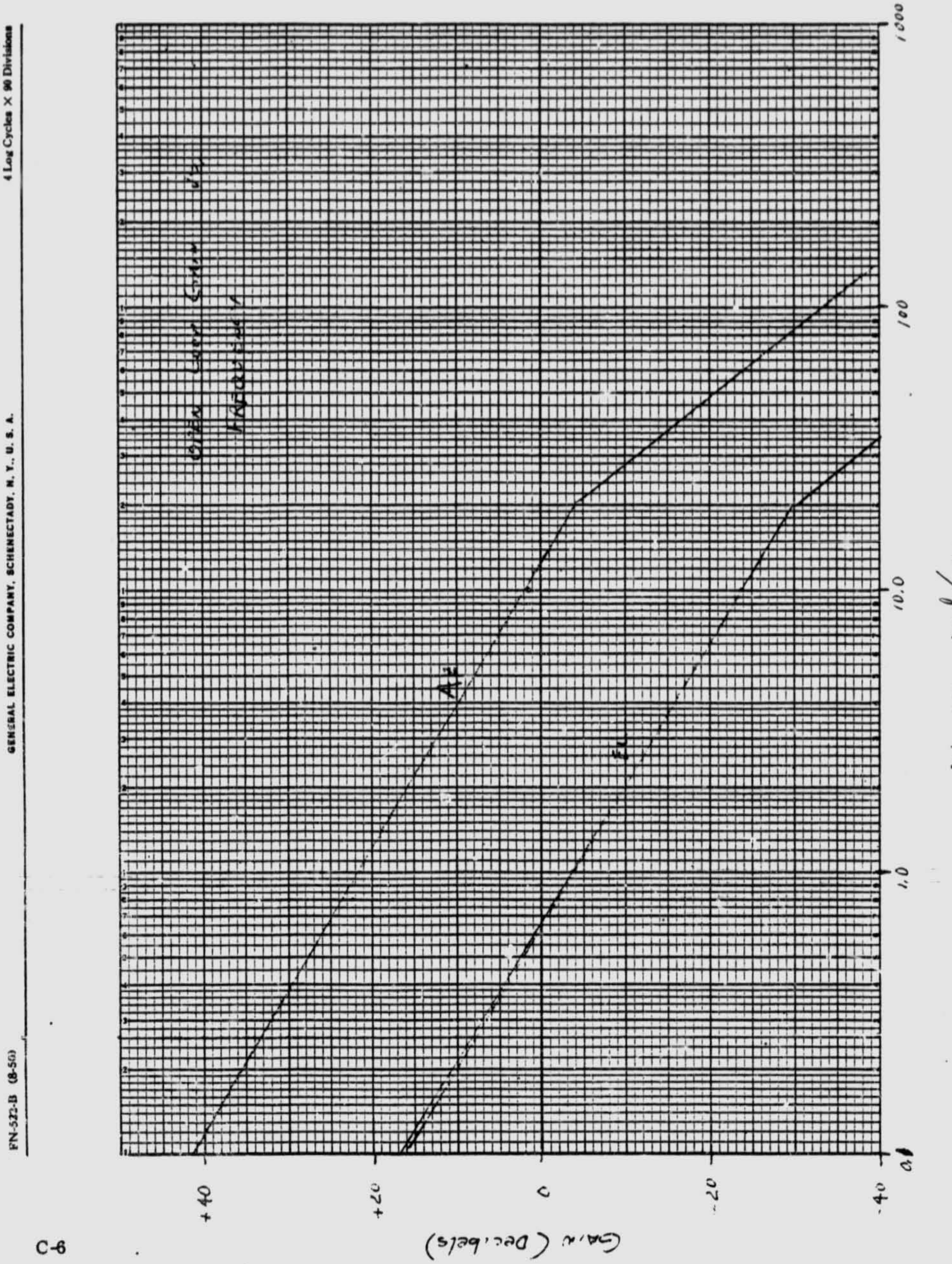


Note that the above values of torque gain represent the <u>minimum</u> values so as to reduce steady state errors to acceptable levels. Actual gain levels can be increased so as to provide a faster slew response. It is proposed that the forward gain in both azimuth and elevation be set at 1.2 ft. lbs./degree. From the attenuation plots this would appear to be about the maximum acceptable value without incorporating compensation.

Based on the above the angular errors due to finite closed loop gain become:

Azimuth = 0.5°

Elevation = 0.027°

Note that a gain of 1.2 ft. lbs./degree will result in about 1/3 of peak available torque (with no heat sink)of 336 lbs. for angular rotations of about 100°.

Any further increase in gain may result in further need for stabilization. Such an increase might also result in a need for a tachometer feedback so as to prevent the load from exceeding a 120°/min angular rate.

Note that for the same gains the angular errors at 1g can be expected to be about 50 times the errors listed above. It is proposed that forward gains be increased so as to provide more realistic performance at 1g.

Tentatively an error budget as suggested:

	At 0g		At 1g	
Error Source	Az	El	Az	El
Due to finite gains in feedback loop	0.5°	0.027°	5°*	0.27*
Gear backlash	0.031	0.036	0.031	0.036
Backlash and non-linearities in potentiometers	0.02	0.02	0.02	0.02

*Assumes forward gain increased by factor 10 at 1g.

Concluding Note

This is a non-linear servo problem which does not readily admit of a satisfactory solution in an examination as brief as this. It is recommended that a more thorough evaluation be made by methods such as "describing function analysis" or preferrably by simulation.

APPENDIX D

EXECUTIVE SUMMARY

VU-GRAPHS

FLEXIBLE INSTRUMENT MOUNT FOR SPACELAB

- DEFINITION
- THE FLEXIBLE INSTRUMENT MOUNT (FIM) IS A 2-AXIS GIMBALLED MOUNT SIZED TO ACCOMMODATE LARGE SPACELAB PAYLOAD INSTRUMENTS
- THE FIM ALLOWS OFF-SET POINTING OF INSTRUMENTS USING THE ORBITER AS A POINTING PLATFORM
- THE FIM INCREASE SPACELAB MISSION FLEXIBILITY BY ALLOWING SEVERAL INSTRUMENTS TO POINT IN DIFFERENT DIRECTIONS SIMULTANEOUSLY, USING THE ORBITER POINTING CAPABILITIES.

NOTE: FIM IS NOT A POINTING SYSTEM

FIM STUDY

- STUDY OBJECTIVES
  - DEFINE CONCEPTS FOR A FLEXIBLE INSTRUMENTS MOUNT FOR SPACELAB
  - SELECT THE MOST PROMISING CONCEPT AND CONDUCT AN ENGINEERING DEFINITION STUDY
  - GENERATE A FIM DEVELOPMENT PLAN
  - GENERATE PRELIMINARY SYSTEM COST.

FIM REQUIREMENTS

- INSTRUMENT ACCOMMODATION
  - SIZE: 2m x 2m x 3m LENGTH, and 1.2 to 2m dia. x 3m LENGTH

    FIM COMPLIANCE - 2.25m dia. x 2.5m LENGTH, SUFFICIENT TO ACCOMMODATE 24 LAMAR MODULES
  - WEIGHT: 950 to 2000 Kg

    FIM COMPLIES
  - STRUCTURAL INTERFACE: CENTER SUPPORT FLANGE

    FIM COMPLIES
  - CG - OFFSET: 0.25m in 1-G

    FIM COMPLIANCE = 0.05m WITHOUT GSE, LARGER OFF-SET REQUIRES GSE

FIM REQUIREMENTS (CONTINUED)

- INSTRUMENT SERVICES
  - POINTING RANGE: 60° HALF ANGLE CONE AROUND INSTRUMENT CENTER LINE
    FIM COMPLIES
  - POSITION ACCURACY: 1° RELATIVE TO FIM BASE
    FIM COMPLIES
  - POSITION READOUT: 0.1°
    FIM COMPLIES
  - SLEWING MANEUVERS: ONCE EVERY 10 MINUTES (MAX)
    FIM COMPLIES
  - SLEW RATE: 120°/min max. (goal)
    40°/min min.
    FIM COMPLIES
  - ELECTRICAL SERVICES: HARNESS ACROSS GIMBALS TO CONNECT INSTRUMENT TO SPACELAB EPDB AND RAU
    FIM COMPLIES
  - THERMAL CONTROL: ACCOMMODATE THERMAL CANNISTER
    FIM COMPLIES

FIM REQUIREMENTS (CONTINUED)

- SHUTTLE/SPACELAB INTERFACES
  - ENVELOPE: STAY WITHIN ORBITER CARGO BAY ENVELOPE DURING ALL ON-ORBIT OPERATIONAL PHASES TO AVOID REQUIREMENT FOR EMERGENCY JETTISON CAPABILITY
    FIM COMPLIES (INSTRUMENT LENGTH CONSTRAINT)
  - STRUCTURAL: USE SPACELAB PALLET HARDPOINTS OR ORBITER KEEL AND TRUNNION FITTINGS
    FIM COMPLIES (USES PALLET HARDPOINTS)
  - THERMAL: CONTROL FIM BULK TEMPERATURE TO $20^{o}C \pm 20^{o}C$
    FIM COMPLIES (PASSIVE THERMAL CONTROL)
  - ELECTRICAL: USE SPACELAB ELECTRICAL POWER AND COMMAND AND DATA MANAGEMENT SYSTEM
    FIM COMPLIES
- OPERATIONS
  - CONTROL: ON-ORBIT PAYLOAD SPECIALIST AND POCC CONTROL
    FIM COMPLIES (USES SPACELAB CDMS)
  - DESIGN LIFE: 10 YEAR OR 50 MISSION
    FIM COMPLIES (WITH PERIODIC MAINTENANCE AND REFURBISHMENT)
  - INTEGRATION AND TEST: 1-G TESTING WITHOUT SPECIAL, GSE
    FIM COMPLIES

FIM CONCEPT IDENTIFICATION

- BASIC FIM ALTERNATIVES
  - AXIS ORIENTATION

    A. AZIMUTH - ELEVATION
    B. ROLL - PITCH

  - SHUTTLE MECHANICAL INTERFACE

    A. SPACELAB PALLET MOUNT
    B. DIRECT ORBITER MOUNT

  - INSTRUMENT MECHANICAL INTERFACE

    A. CENTER FLANGE
    B. END FLANGE OR PLATE

  - DEPLOYMENT

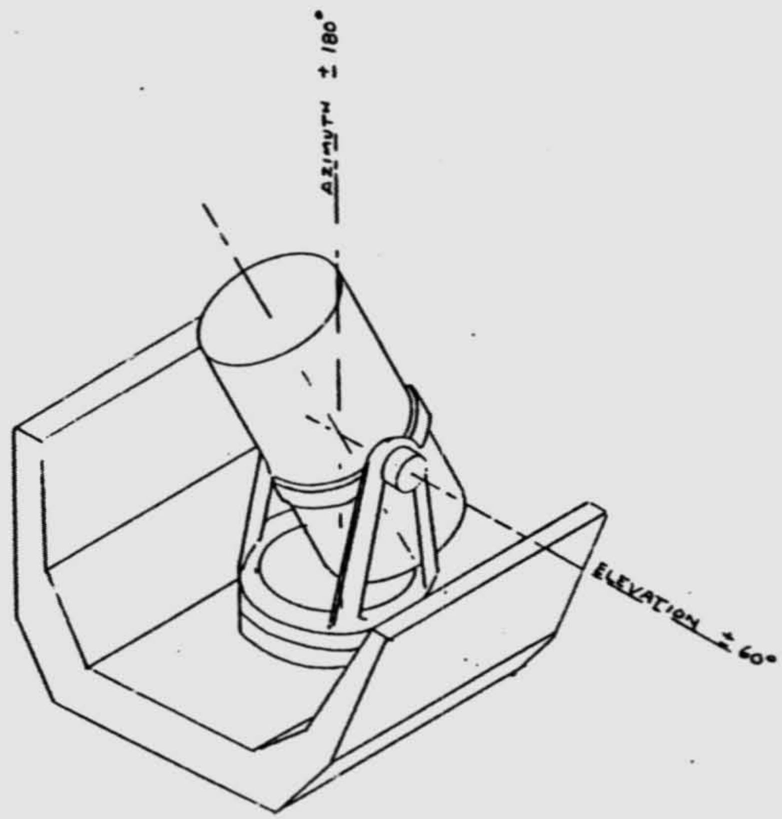
    A. ROTATION ONLY
    B. TRANSLATION
    C. EXTENDABLE SUPPORT

FIM CONCEPT IDENTIFICATION (CONTINUED)

- SELECTED POTENTIAL FIM CONCEPTS
  - CONCEPT #1: AZIMUTH - ELEVATION AXES, CG - MOUNT
  - CONCEPT #2: ROLL - PITCH AXES, CG - MOUNT
  - CONCEPT #3: AZIMUTH - ELEVATION AXES, END - MOUNT
  - CONCEPT #4: AZIMUTH - ELEVATION AXES, END - MOUNT, DEPLOYABLE
  - CONCEPT #2A: ROLL - PITCH AXES, CG - MOUNT (EXTENDED)
  - CONCEPT #4A: AZIMUTH - ELEVATION AXES, CG - MOUNT, NON-PALLET MOUNTED, DEPLOYABLE.

FIM CONCEPT #1

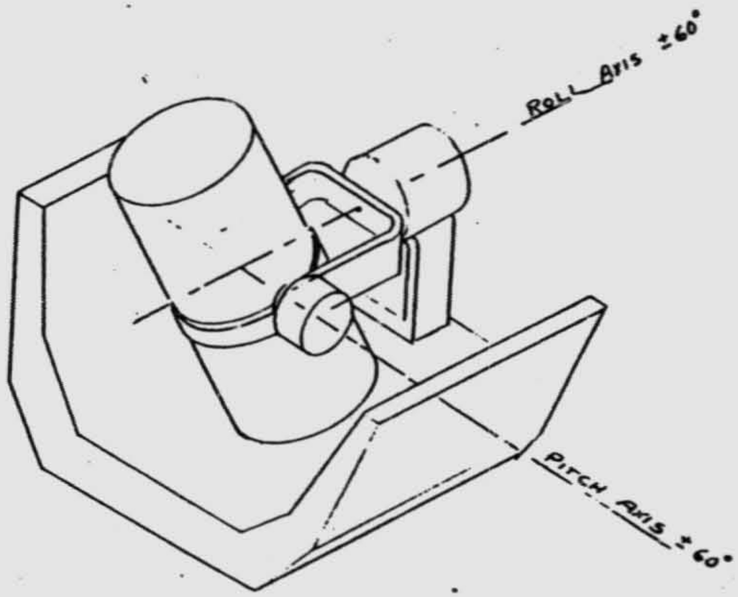
- AZIMUTH - ELEVATION AXES, CG - MOUNT



NOTE: SELECTED CONCEPT

FIM CONCEPT #2

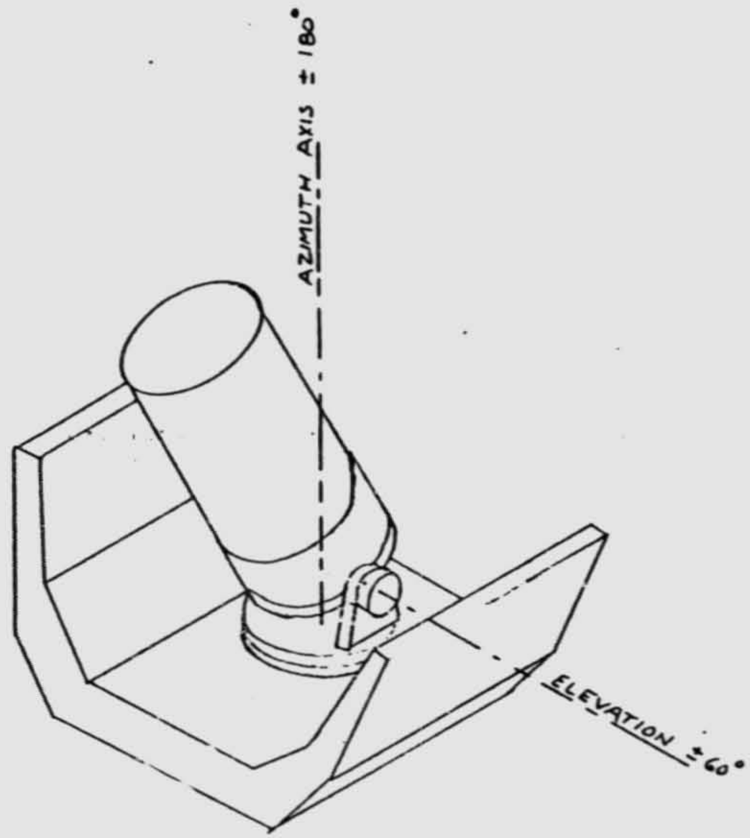
- ROLL-PITCH AXES, CG - MOUNT



NOTE: PALLET OVERHANG FOR FULL 60° FOV
MORE COMPLEX POINTING CONTROL THAN AZIMUTH-ELEVATION MOUNT.

FIM CONCEPT #3

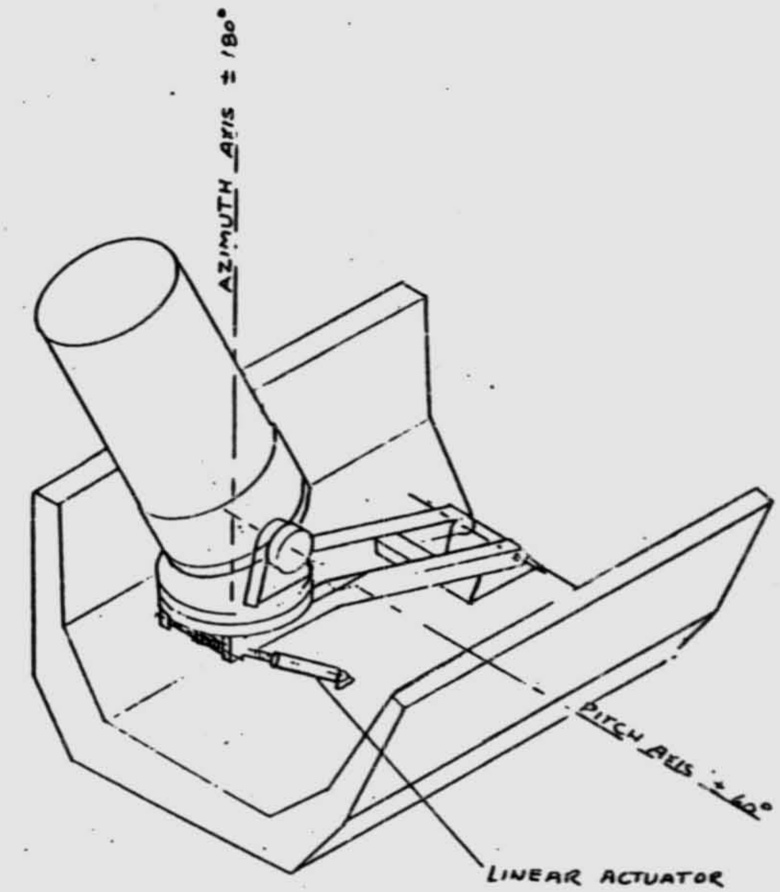
- AZIMUTH - ELEVATION AXES, END - MOUNT



NOTE: NOT A CG - MOUNT

FIM CONCEPT #4

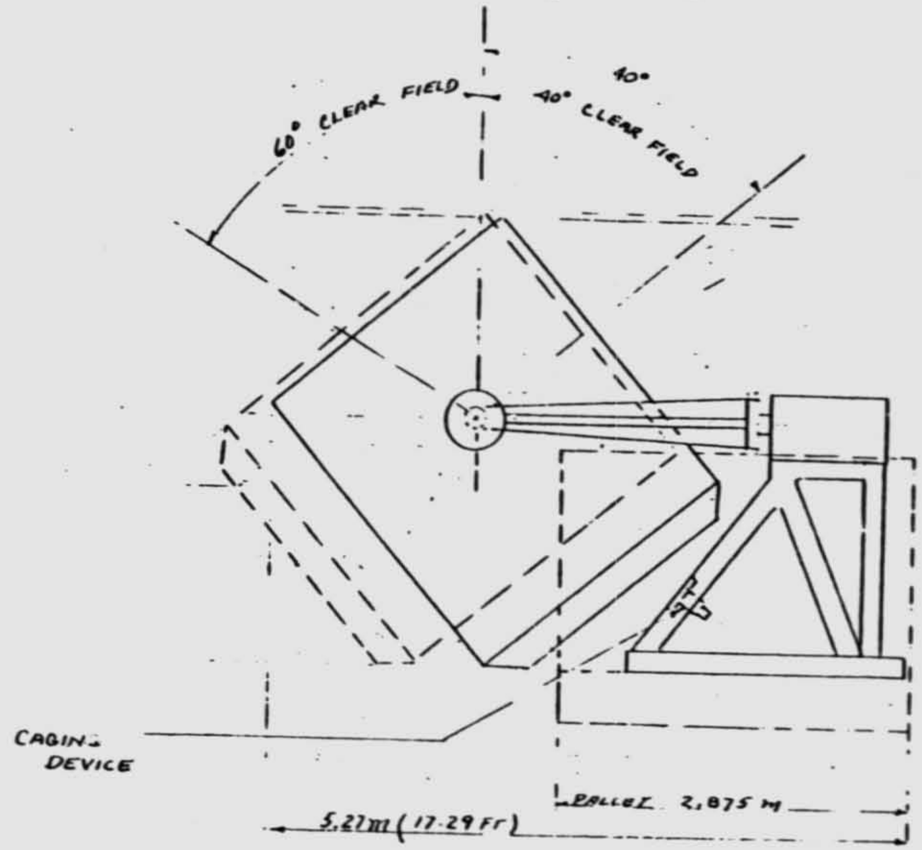
- AZIMUTH - ELEVATION AXES, END - MOUNT, DEPLOYABLE



NOTE: PENETRATES (NOT A CG - MOUNT ORBITER) CARGO BAY ENVELOPE REQUIRES EMERGENCY JETTISON CAPABILITY

FIM CONCEPT #2A

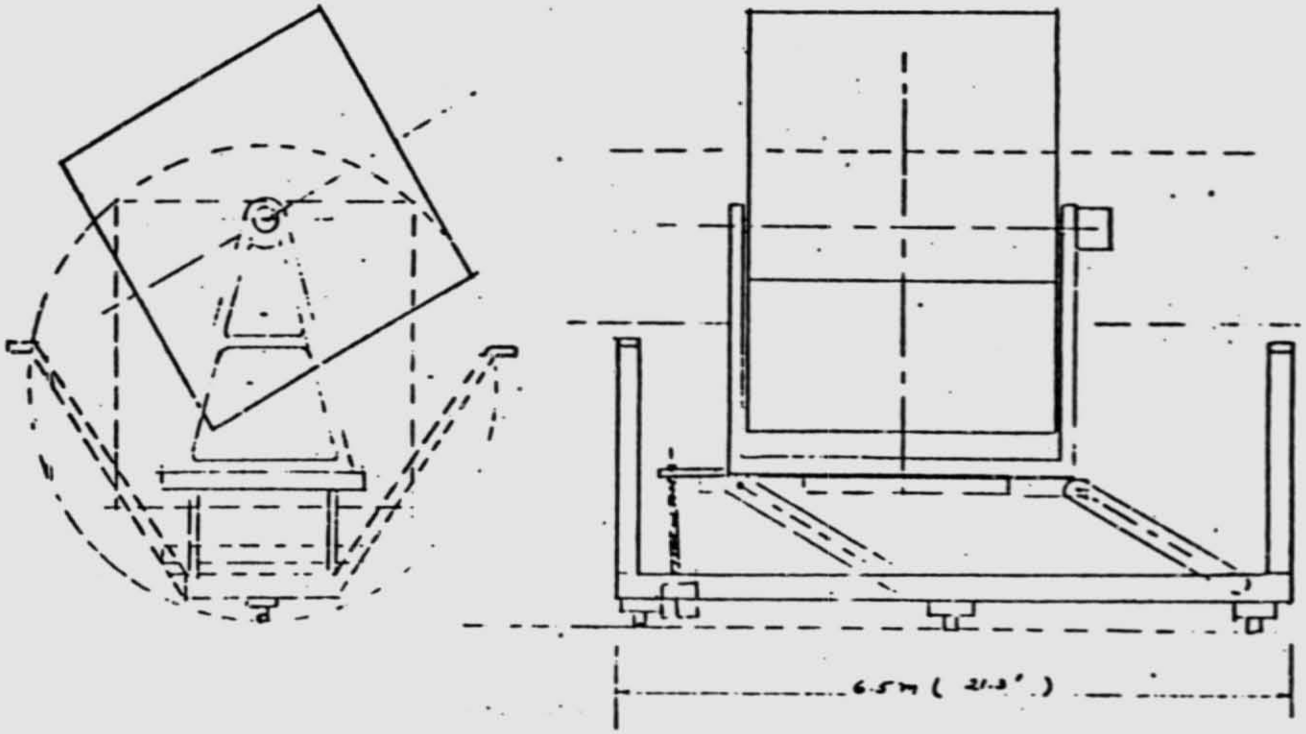
• ROLL - PITCH AXES, CG - MOUNT, EXTENDED



NOTE: LARGE PALLET OVERHANG
PENETRATES ORBITER CARGO BAY ENVELOPE

FIM CONCEPT #4A

• AZIMUTH - ELEVATION AXES, CG - MOUNT, NON-PALLET MOUNTED



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NOTE: REQUIRES SEPARATE SUPPORT STRUCTURE FOR ORBITER CARGO BAY MOUNTING
REQUIRES ALMOST TWO PALLET LENGTH
PENETRATES ORBITER CARGO BAY ENVELOPE

FIM CONCEPT EVALUATION

- IMPORTANT EVALUATION CRITERIA

  - USE OF SPACELAB PALLET FOR FIM MOUNTING

    ACCOMMODATION OF PAYLOAD SIZES, i.e., 24 LAMAR MODULES AND CYLINDRICAL PAYLOADS OF 2 m DIAMETER AND 3 m LENGTH.

    MINIMUM REQUIRED LENGTH IN THE ORBITER CARGO BAY, i.e. MAXIMUM UTILIZATION OF THE SPACELAB PALLET VOLUME AND MOUNTING AREA.

    STAYING WITHIN THE ORBITER CARGO BAY PAYLOAD ENVELOPE DURING ALL ON-ORBIT OPERATIONAL PHASES TO AVOID EMERGENCY JETTISON REQUIREMENTS.

    FULL 60° FIELD OF VIEW

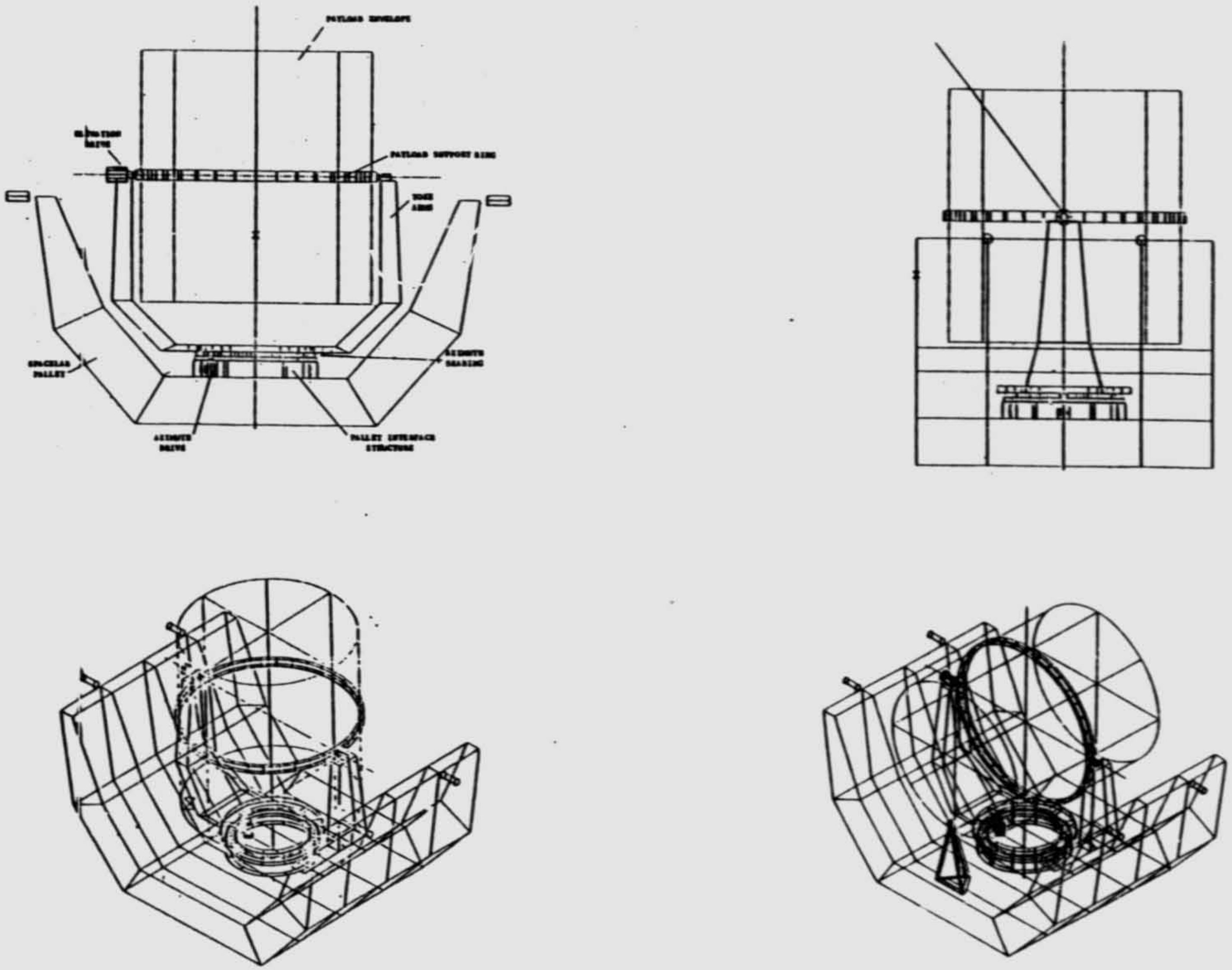
    PROVIDE INSTRUMENT INTERFACE AT AN INSTRUMENT CENTRAL SUPPORT RING.

    GROUND TESTING WITHOUT EXTENSIVE NEED FOR GSE.

    SIMPLE AND STANDARD INTERFACES LIMITED TO SPACELAB.

CONCEPT #1 MET MOST OF THE EVALUATION CRITERIA AND WAS SELECTED FOR FURTHER ENGINEERING EVALUATION

SELECTED FIM CONCEPT



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FIM CHARACTERISTICS

- WEIGHT - 850 Kg
- POWER - 70 WATTS MAX. AT MOTOR STALL
  35 WATTS FOR SLEWING
  3 WATTS FOR POSITION HOLDING IN ORBIT
- PAYLOAD SIZE - CYLINDER, 2.25 m.dia. x 2.5 m LENGTH
  CENTRAL PART OF THE CYLINDER CAN BE LONGER
  THIS ACCOMMODATES 24 LAMAR MODULES AND THE GAMMA-RAY SPECTROMETERS IDENTIFIED FOR THIS STUDY
- PAYLOAD WEIGHT - 2000 Kg
- ERROR BUDGET (IN ORBIT) - AZIMUTH 0.5^{o}
  ELEVATION 0.1^{o}
- CG-OFFSET - 5 m ALLOWED ON GROUND WITHOUT GSE
  SIGNIFICANTLY MORE ALLOWED ON ORBIT
- FIM STRUCTURE FREQUENCY - 11 Hz WITHOUT CAGING, HIGHER WITH CAGING
  (SPACELAB PALLET "DESIRE": 25 Hz)
- PALLET INTERFACE - 6 PALLET HARDPOINTS, PRELIMINARY ANALYSIS INDICATES THAT HARDPOINT LOADING MIGHT BE WITHIN ALLOWABLE LIMITS.

SELECTED FIM CONCEPT

- FIM CONCEPT
  - AZIMUTH - ELEVATION AXES
  - CG - MOUNT
  - PALLET MOUNTED, NO PALLET OVERHANG
  - STAYS WITHIN ORBITER CARGO BAY ENVELOPE
- FIM SYSTEM COMPONENTS
  - STRUCTURE SUBSYSTEM - PAYLOAD INTERFACE RING
    YOKE ARMS
    BOTTOM RING STRUCTURE
    AZIMUTH BEARING
    ELEVATION (PITCH) BEARING
    PALLET INTERFACE FRAME
    CAGING DEVICE
  - ELECTROMECHANICAL DRIVE SUBSYSTEM - TORQUER METERS
    HARMONIC SPEED REDUCER
  - ELECTRONIC CONTROL SUBSYSTEM - POSITION ENCODER
    OTHERS (TBD)

AREAS REQUIRING FURTHER STUDY

- FIM/PALLET INTERFACE - FIM/PALLET COUPLED LOAD ANALYSIS
- FIM/PAYLOAD CAGING DESIGN
- INTERFACE BETWEEN AZIMUTH BEARING AND FIM/PALLET INTERFACE STRUCTURE
- FIM CONTROL SYSTEM DESIGN/ANALYSIS
- FIM THERMAL CONTROL

FIM DEVELOPMENT PLAN

- PHASE C/D WBS GENERATED
- PHASE B WBS GENERATED
- "MODULAR" PHASE B APPROACH DEVELOPED TO MATCH A FIM DEFINITION PROGRAM WITH EXISTING FUNDING CONSTRAINTS
- TASKS IDENTIFIED WHICH NEED TO BE CONDUCTED IN MODULAR PHASE B APPROACH
- NON-CRITICAL PHASE B ACTIVITIES POSTPONED UNTIL PHASE C/D

FIM PHASE B SCHEDULE